

International Marine Engineering

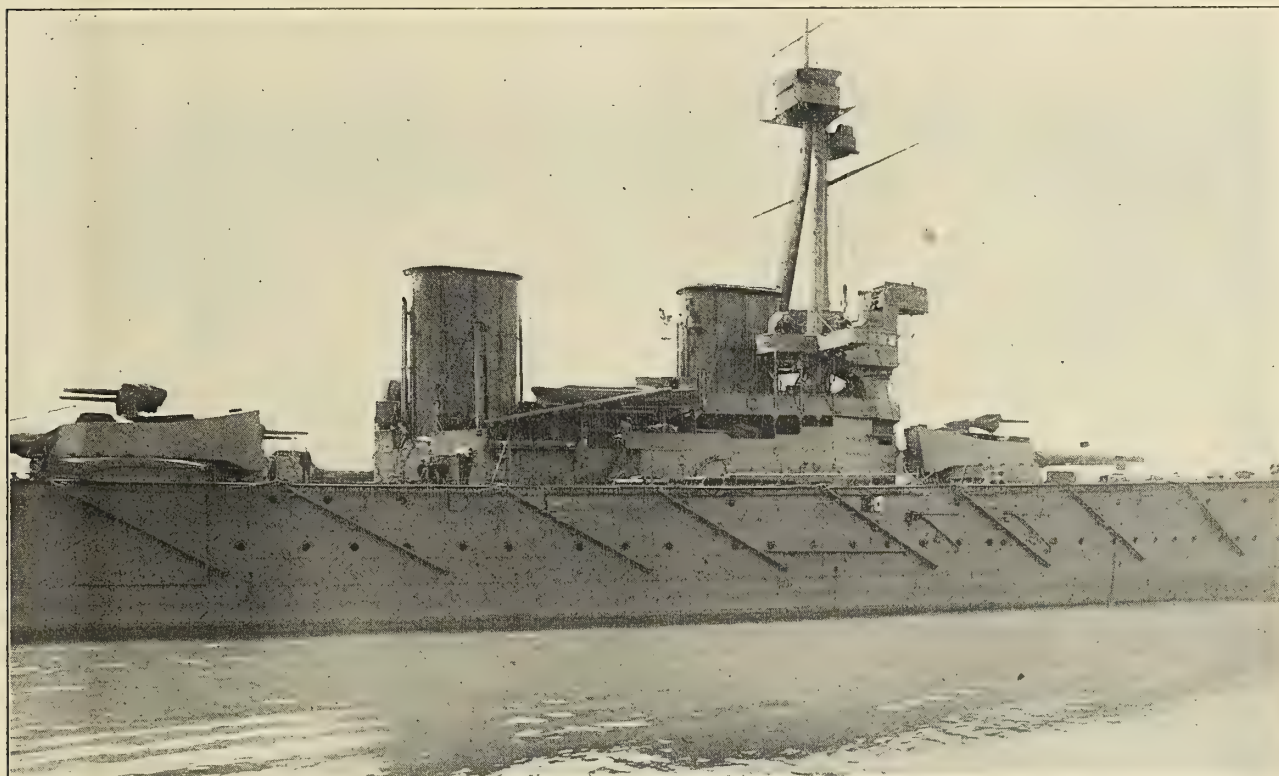
AUGUST, 1908.

BRITAIN'S NEW TURBINE BATTLE CRUISER INDOMITABLE.

The new *Indomitable*, which has created a world-wide interest, has now passed through all her official steam and gun trials with the greatest success. During her 24-hour speed test she is said to have averaged the high speed of 26.75 knots, and at times actually reached 28 knots. The contract speed was 25 knots. Following out the conditions of secrecy which have been observed during the construction of this ship, it has not been considered advisable to give any particulars of the speed trials; but we can now deal more freely with the "ship of mystery" than was the case when we gave a description of

greatest length of any warship afloat or projected), with a breadth of 78 feet 6 inches, and displaces 17,250 tons, on a draft of 26 feet. Her side armor is 7 inches in thickness amidships, tapering to 4 inches at bow and stern. The gun barbettes are of 8-inch armor, while the turrets are of 7-inch armor.

We give reproductions of several photos of the ship, clearly illustrating her appearance and fighting power. It will be noted that our last description of the position and number of the main guns was correct. There are eight 12-inch guns,



THE CENTRAL PORTION—STARBOARD SIDE—OF THE BRITISH CRUISER-BATTLESHIP INDOMITABLE.

the vessel in our issue of May, 1907, because she has now passed from the builder's closely-guarded private works to the open sea, where she is unable to be concealed from the keen, searching eye of the photographic camera. The *Indomitable* is the first completed of the three ships of the *Invincible* class, having been constructed in the record time of two years and two months. This gives some indication of the high standard attained in shipbuilding on the Clyde.

The main features of the ship were described last year, but one or two facts may be repeated here. The vessel is 530 feet long between perpendiculars (560 feet over all, this being the

each of 45 calibres in length, firing an 850-pound shot, with a muzzle energy of 47,697-foot tons. The forward pair of 12-inch guns is on the forecastle deck, and the four amidship guns are on the same deck level, and situated between the second and third funnels, as shown in illustrations. The after pair of guns is on the upper deck, apparently about 9 feet lower than the forecastle deck. All the eight guns are arranged to fire on either broadside, and six guns may be fired ahead and a similar number astern, so that in gun power the *Indomitable* is equal to the battleship *Dreadnought*. The secondary armament consists of a large number (sixteen) of



VIEW FROM OFF THE STARBOARD QUARTER OF THE CRUISER-BATTLESHIP INDOMITABLE.

4-inch guns, half of them mounted in pairs on the four turrets. This arrangement permits the training of twelve of these guns on either broadside.

To the outside observer the ship has many new features. A very noticeable detail is the arrangement of the masts. They are both of the tripod type; this entirely dispenses with the shrouds, which would have obstructed the training of some of the guns. Also, the tripod form reduces the possibility of the mast's being completely shot away, for even if one leg did go the others would stand, with temporary guy ropes. This is an important detail, because the masts carry the gun sighting ("fire control") stations and the receiving wires in connection with the wireless telegraph system. The customary gaff for the aerial wire has been dispensed with, and the masts have been increased in height so as to give an aerial wire

arrangement between the masts in a manner similar to that adopted on the *Lusitania*.

Then there is almost an entire absence of boat davits, as most of the boats have been placed so that they may be put overboard with derricks. But the ship's lifeboats are very conspicuous, in long goose-neck davits, so arranged that the boat can be lowered over the ship's side or stowed inboard clear of gun fire. Another noticeable detail is the position of the compass platform, which is very high up, and appears to be almost level with the top of the funnel. The ship is also provided with torpedo-net defense, all fore and aft.

The propelling turbines are of the Parsons type, designed to give 41,000 horsepower, and are placed in two compartments, divided by a center-line bulkhead. They are similar to those in the *Dreadnought*. Thus, in each engine room there



THE CRUISER-BATTLESHIP INDOMITABLE HAS THE GREATEST LENGTH OF ANY WARSHIP AFLOAT.

are one high-pressure and one low-pressure main ahead turbine, a cruising turbine, and one high-pressure and one low-pressure astern turbine. The high-pressure ahead and the high-pressure astern drive the outer shafts, while the cruising and the low-pressure ahead and low-pressure astern are on the inner shafts. The four shafts have one propeller on each, and these turn outboard when going ahead. The two outer propellers are placed forward of the inner ones. The ship has two rudders of the balanced type, which are hung from the stern structure. They are similar to those in the *Dreadnought*, and, being of larger area, give the ship a great turning power.

MARINE ENGINE DESIGN.

BY EDWARD M. BRAGG, S. E.

STRENGTH OF MATERIALS.

The cylinder diameter and cut-offs having been determined, we are in a position to approximate to the loads coming upon the different parts, and to proportion them so that they shall be strong enough to carry these loads. There are a great many formulæ in use for this purpose, but many of them are in such shape that they cannot be readily adapted to all conditions. Most of them are based upon the assumption that steel of a certain ultimate strength is to be used, and that, therefore, the allowable working stress is a fixed quantity. Since, however, the ultimate strength of the steel used in marine engines varies from 60,000 pounds to 100,000 pounds per square inch, it is much better to base the calculations upon the allowable factor of safety, and that is the principle which will be followed in this system of design.

The factor of safety which it is allowable to use in the design of a given part depends upon the kind of load to which it is subjected, the construction of the part itself, and upon the conditions under which it works. There are three kinds of loads: steady loads, intermittent loads and alternating loads. The steady load is one which is applied originally in an appreciable length of time, and the stress resulting in either tension or compression of constant amount. The intermittent load is one which is applied more or less suddenly, and produces either tension or compression; the stress varying from zero to a maximum, or from a minimum to a maximum. The alternating load is of such a nature as to cause the stress to change alternately from tension to compression.

The elastic limit is the limit of the stress to which a material may be subjected, and have the strain or deformation proportional to the stress. The elastic limit is not the same under all conditions. When the steel comes from the mill, it may have an unnatural elastic limit, due to its treatment during manufacture. When subjected to a moderate alternating load for a short time, the effect of this treatment will be overcome, and the piece will have more nearly its natural elastic limit. The tension elastic limit may be increased by subjecting it to successively increasing loads, but its elastic limit in compression will be correspondingly decreased; so that a piece which has been subjected to large intermittent tension loads might fail at a comparatively small load in compression. The elastic limit in compression can be restored by subjecting the piece to successively increasing compression loads, the elastic limit in tension then being correspondingly decreased.

It has been found by experiment that a piece which would carry a constant total load of 80,000 pounds tension for an indefinite time, would fail after a while if the load were an intermittent one varying from 80,000 pounds to 40,000 pounds; it would fail sooner if the load varied from 80,000 pounds to zero; and would fail in a comparatively short time if subjected to an alternating load varying from 80,000 pounds in tension to 80,000 pounds in compression. In order to have the piece

equally strong under each condition of loading, the maximum of 80,000 pounds constant load which the piece will carry indefinitely should be reduced to 40,000 pounds if the load is intermittent, and varies from 40,000 pounds to zero, and to something between 40,000 pounds and 80,000 pounds if the minimum is 30,000 pounds instead of zero. If the stress varies from tension to compression, the load should vary from not more than 27,000 pounds tension to 27,000 pounds compression, in order that the piece may last indefinitely. In other words, the destructiveness of the three kinds of loads, constant, intermittent and alternating, is about in the ratio of 1, 2, 3, respectively.

Experiment has shown that for steel the elastic limit is from one-half to two-thirds of the ultimate strength. It is customary to choose the working stress so that it shall be less than the elastic limit of the material, usually less than one-half of it. It results, therefore, that in some structural work on land, the factors of safety employed are 3, 6 and 9 for steady, intermittent and alternating loads.

The statement was made above that the conditions under which the parts work affect the factors of safety. The breaking

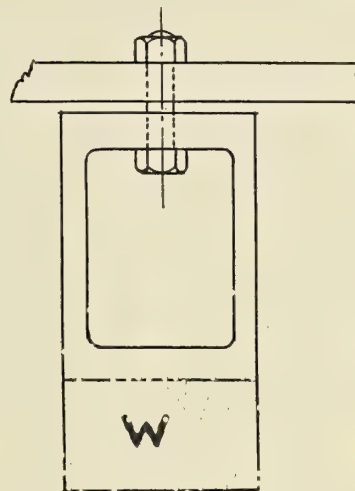


FIG. 4.

of the parts of an engine in a ship would be attended with so much danger, and the facilities for repair are so limited, that somewhat larger factors of safety should be employed for marine engines than for structural work on land. In this system of design, the factors of safety used for engines of the merchant marine are 4, 8 and 12 for steady, intermittent and alternating loads, respectively.

In engines for the vessels of the navy, the factors of safety employed are more nearly 3, 6 and 9. The reasons for this are that it is desirable to save weight, the life of the engine is comparatively short, and the engines are seldom run at full power. Naval vessels are usually cruising around at one-half or two-fifths power, so that although the factors of safety are smaller for full power, they are amply large for the reduced power which the engines develop during the greater part of their existence.

Practically there are no parts of a marine engine which can be designed for steady loads. Some of the bolts and threaded parts may be subjected to steady loads if the nuts are set up tight enough, but it would not be safe to count on the nuts always being as tight as when they were first set up. The stress in the body of a bolt is always that due to setting up the nut, unless the load upon the bolt is in excess of this. Referring to Fig. 4, it can be seen that the stress in the body of the bolt will be that due to the load *W* as long as the flanges are apart. When the nut is set up sufficiently to bring the flanges together the stress will be increased, and the body of

the bolt will not be stressed further until sufficient weight has been added to W to cause the flanges to separate again. In this way, even with an intermittent load acting upon the parts joined by the bolt, the body of the bolt might be subjected to a steady load if the nut were set up tight enough. In the first place, to subject the bolt to a stress greater than the intermittent load. It would not be advisable to count on this, however, so that a factor of safety as low as 4 should never be used in any part of a marine engine. The lowest factor of safety that we shall use will be 8 for merchant engines and 6 for naval engines.

The parts subjected to intermittent loads are bolts, the threaded portion of piston rods and certain portions of valve stems. In the case of bolts and other threaded parts, a deviation from the chosen factor of safety 8 is made, because of the construction of the part. A piece of steel which has been nicked will fail at a lower stress, the net area at the nick being used, than if the same net area were in a piece without a nick. The sharp angle at the root of the threads has the same effect upon the bolt as the nick just referred to, so that the metal in the threaded portion cannot be stressed as greatly as it could be if the surface of the bolt were perfectly plain. The factor of safety to be used for bolts will be increased on this account to 10 for merchant engines and to 8 for naval engines.

Another condition which causes a further modification of the factor of safety for bolts is the initial torsional stress set up when the nut is put on, due to the friction between the threads of the nut and bolt. This initial stress is greater relatively in bolts of small diameter than in bolts of larger diameter, and can be considered as negligible in bolts of 3 inches diameter and above; so that, starting with a factor of safety of 10 for bolts of 3 inches diameter and above, it will be gradually increased until it is 16 for bolts of 1 inch diameter. In bolts for naval engines, the factor of safety will vary from 8 for bolts of 3 inches diameter and above to 14 for 1-inch bolts. Below is given a table of the factors of safety, areas and allowable loads upon bolts of various diameters, upon the assumption that steel of 60,000 pounds per square inch ultimate strength is used. For steel of higher strength, the allowable load is increased in proportion. It is customary to use not less than 4 threads per inch for the bolts of connecting rods, caps and main bearings and for the threaded portion of the piston rod.

When hexagonal nuts and heads are used, they are usually

TABLE IV.

Diameter of Bolt.	No. of Threads per Inch.	Area at Root of Thread.	MERCHANT.		NAVAL.	
			Factor of Safety.	Allowable Load in Pounds.	Factor of Safety.	Allowable Load in Pounds.
3/4	10	.302	16.70	1,090	14.70	1,240
7/8	9	.420	16.40	1,540	14.40	1,755
1	8	.550	16.00	2,060	14.00	2,355
1 1/8	7	.694	15.60	2,670	13.60	3,065
1 1/4	7	.893	15.25	3,520	13.25	4,060
1 3/8	6	1.057	14.90	4,250	12.90	4,910
1 1/2	6	1.295	14.50	5,360	12.50	6,225
1 5/8	5 1/2	1.515	14.10	6,450	12.10	7,525
1 3/4	5	1.746	13.75	7,625	11.75	8,920
1 7/8	5	2.051	13.40	9,170	11.40	10,800
2	4 1/2	2.302	13.00	10,650	11.00	12,600
2 1/4	4	3.023	12.25	14,800	10.25	17,700
2 1/2	4	3.719	11.50	19,400	9.50	23,500
2 3/4	4	4.62	10.75	25,700	8.75	31,600*
3	4	5.63	10.00	33,800	8.00	42,250
3 1/4	4	6.73	10.00	40,400	8.00	50,500
3 1/2	4	7.90	10.00	47,400	8.00	59,250
3 3/4	4	9.21	10.00	55,200	8.00	69,000
4	4	10.60	10.00	63,500	8.00	79,350
4 1/4	4	12.10	10.00	72,500	8.00	90,700
4 1/2	4	13.68	10.00	82,100	8.00	102,800
4 3/4	4	15.36	10.00	92,100	8.00	115,200
5	4	17.20	10.00	103,000	8.00	129,000
5 1/2	4	21.05	10.00	126,400	8.00	157,500
6	4	25.30	10.00	152,000	8.00	190,000
6 1/2	4	30.00	10.00	180,000	8.00	225,000
7	4	35.00	10.00	210,000	8.00	262,500

of standard dimensions, but a great many cylindrical nuts and heads and collar nuts are used. For cylindrical nuts, $d = 1.75 D$ to $1.67 D$, and $h = D$, where d = diameter of the nut, and h its height. D = diameter of the bolt.

For cylindrical bolt heads: $d = 1.5 D$; $h = 0.67 D$.

The ultimate strength of the materials used in marine engines will be assumed as follows:

Cast iron.....	20,000 pounds (tension).
Cast steel.....	55,000 to 70,000 pounds.
Wrought steel.....	60,000 to 100,000 pounds.
Phosphor bronze.....	35,000 pounds.
Manganese bronze.....	50,000 pounds.

We give a table of collar nuts, where the flange of the part joined is counterbored to serve as a collar for the nut.

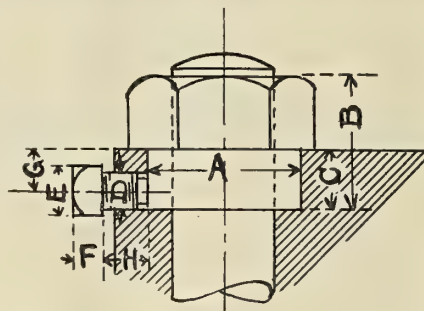


FIG. 5. DIMENSIONS IN INCHES.

TABLE V.

Diameter.	A	B	C	D	E	F	G	H*	H†
1/2	13/16	7/8	7/16	1/4	3/8	1/4	5/16	5/16	3/8
5/16	7/8	7/8	7/16	1/4	3/8	1/4	5/16	5/16	3/8
3/8	1	1	7/16	1/4	3/8	1/4	5/16	5/16	3/8
3/4	1 1/8	1 1/8	9/16	5/16	7/16	5/16	3/8	3/8	7/16
7/8	1 3/8	1 3/8	9/16	5/16	7/16	5/16	3/8	3/8	7/16
1	1 5/8	1 5/8	5/8	3/8	1/2	5/16	7/16	7/16	1/2
1 1/8	1 7/8	1 7/8	5/8	3/8	1/2	5/16	7/16	7/16	1/2
1 1/4	2	2	3/4	7/16	5/8	3/8	1/2	1/2	5/8
1 1/2	2 1/8	2 1/8	3/4	7/16	5/8	3/8	1/2	1/2	5/8
1 3/4	2 3/8	2 3/8	7/8	3/2	1 1/16	7/16	9/16	1/2	5/8
1 5/8	2 5/8	2 5/8	7/8	3/2	1 1/16	7/16	9/16	1/2	5/8
1 7/8	2 7/8	2 7/8	7/8	3/2	1 1/16	7/16	9/16	1/2	5/8
2	3	3	1	1	1 1/2	1 1/2	1 1/2	1 1/2	3/4
2 1/2	3 1/2	3 1/2	1 1/8	1 1/8	1 3/4	1 3/4	1 3/4	1 3/4	3/4
3	4	4	1 1/4	1 1/4	2	2	2	2	1
4	6	6	1 3/4	1 3/4	2 3/4	2 3/4	2 3/4	2 3/4	1 1/4
5	7 1/2	7 1/2	2	2	3 1/2	3 1/2	3 1/2	3 1/2	1 1/2
6	9	9	2 1/4	2 1/4	4	4	4	4	1 3/4

* Wrought iron and composition. † Cast iron.

Sometimes the collar is a separate piece attached to the flange by pins or dowels.

CYLINDERS.

The cylinders may be divided into two classes, those with liners and those without liners. If the cylinders are to be jacketed, of course liners will be necessary, but very often liners are used when it is not intended that the space between the liner and the cylinder barrel shall be used for jacket steam. The presence of a liner makes it easier to warm up the engine before starting, by temporarily admitting steam to the jacket space. The liners are very simple in shape, and can be made of a tough grade of cast iron, which will give a good wearing surface for the piston rings, while the more complicated cylinder casting can be made of a softer grade, which will flow readily during casting. A further advantage in the use of a liner is that, if any accident happens to the piston, it may break the liner, which can be readily replaced, rather than the barrel of the cylinder, the replacing of which would necessitate the dismantling of a good part of the engine.

In deciding upon whether or not steam jackets are to be used, the following points should be considered: range of

temperature in one cylinder, temperature of steam to be used in jacket spaces and number of revolutions of cranks per minute. The layer of metal in the cylinder walls which is subjected to any great variation of temperature is usually comparatively thin, and the purpose of the steam jacket is to keep this layer as thin as possible. The layers of metal near the outside of the wall are almost constant in temperature, and the temperature of the jacket steam must be considerably higher than the natural temperature of the outside of the wall, in order that its effect may be felt upon the inner layers, and result in any reduction of the amount of metal subjected to variations of temperature. The use of boiler-pressure steam in the jackets of the high-pressure cylinder of triple and quadruple expansion engines has, therefore, very little beneficial effect upon the economy of the engine; and, if the space is not well drained of water, may have a harmful effect. It is quite common to use no steam in the high-pressure jackets, but to use jacket steam on the medium-pressure and low-pressure cylinders only. The element of time will also affect the thickness of the layer of metal subjected to variation of temperature, so that the higher the number of revolutions per minute the less need there is of jacket steam.

The load upon the cylinder walls of the high-pressure and medium-pressure cylinders is intermittent in character, varying from a minimum to a maximum. In the low-pressure cylinder, when a condenser is used, the pressure at times will be less than atmospheric, and consequently the load will be alternating in character. The load to be used in calculating the thickness of the walls will depend upon whether or not jacket steam is to be used. If the jackets are not to be used except for warming up, and reducing valves and relief valves are fitted, so that the pressure in the jackets cannot be greater than that for which they are designed, it may be assumed that the load upon the cylinder walls is as follows:

High-pressure — maximum	= boiler pressure, gage.
Triple; medium-pressure — maximum	= 0.5 boiler pressure.
Quadruple; first medium-pressure — maximum	= 0.6 boiler pressure.
Quadruple; second medium-pressure — maximum	= 0.4 boiler pressure.
Quadruple and triple; low-pressure — maximum	= 0.25 boiler pressure.

The maximum pressure in the low-pressure cylinder will seldom be more than 35 pounds absolute, or 20 pounds gage, and the back pressure seldom less than 4 pounds absolute, so the range of pressure would be 31 pounds, or about one-sixth of the absolute boiler pressure. When this is increased by 50 percent, to allow for the fact that it is an alternating load, we have one-quarter of the absolute boiler pressure as the equivalent load acting upon the low-pressure cylinder walls.

These assumptions are for the engine working under ordinary conditions. There are, however, other conditions which may exist at times, and for which allowance should be made. The high-pressure cylinder is liable to have water carried into it with the steam from the boiler; the medium-pressure and low-pressure cylinders may have their power increased by admitting live steam into the receivers, and so increase the pressure at which these cylinders take steam. Allowance for all these things is generally made by adding a fixed amount to the pressure assumed, so that the formula for barrels and liners is as follows:

$$t = \frac{(P + 25)D}{6,000} + \frac{40}{100 + D} \quad (15)$$

where t = thickness of walls in inches,

P = maximum pressure in cylinder, assumed as above,

and D = diameter of cylinder, as calculated.

Equation (15) may be used for cylinders without liners. When liners are used they may be calculated by (18), and the barrels made of the same thickness. When used for the liner, the term

$$\frac{40}{100 + D}$$

adds something to the thickness, to allow for reborring later, and to insure that the walls shall be of a thickness practical for casting, if the diameter D is small. When (15) is used for the barrel of a cylinder with a liner, the

$$\frac{40}{100 + D}$$

is not needed for reborring, but is required because the barrel is larger in diameter than the liner, and may be thinner in places than designed, due to the displacement of the core during casting. Since the liner is finished inside, and at certain places outside, it will be of the designed thickness.

If jacket steam is to be used, and the pressure of the steam is to be greater than the $(P + 25)$ in the formula, the greater pressure should be used in obtaining the necessary thickness of barrel and liner.

All of the cylinder barrel thicknesses and liner thicknesses are usually made the same. The thickness for each cylinder can be found, and then an average taken, or only the thickness for the high-pressure cylinder need be figured, and all the others made the same. The thicknesses of the other parts of the cylinder and valve chest casting can be made some fraction of the liner or barrel thickness t , thus:

Thickness of cylinder bottom, single,	= t ,
Depth of ribs of cylinder bottom, single,	= $5t$,
Thickness of ribs of cylinder bottom, single,	= $t - \frac{1}{16}$ inch,
Thickness of cylinder bottom, double,	= $0.9t$,
Distance between walls, double,	= $5t$ (at least),
Thickness of cylinder flange,	= $1.3t$ to $1.4t$,
Width of cylinder flange,	= $2.75t$ to $3.25t$,
Thickness of metal in cylinder feet,	= t ,
Thickness of flange on cylinder feet,	= $1.5t$ to $1.75t$,
Diameter of bolts for cylinder feet,	= $1.4t$ to $1.6t$,
Thickness of metal in cylinder cover, single,	= t ,
Thickness of metal in cylinder cover, double,	= $0.85t$,
Spacing of webs in cover and bottom	= $p = \frac{t \times 100}{\sqrt{P}}$,

Thickness of metal in valve liners, = t ,

Thickness of metal in ports and passages = $0.85t$ to $0.9t$.

The clearances are usually as follows:

TABLE VI.

Diameter of Cylinder.	Bottom Clearance.	Top Clearance.
16 to 24 inches	$\frac{1}{4}$ to $\frac{3}{8}$ inch	$\frac{3}{16}$ to $\frac{1}{4}$ inch
24 to 40 inches	$\frac{5}{16}$ to $\frac{1}{2}$ inch	$\frac{1}{4}$ to $\frac{3}{8}$ inch
40 to 60 inches	$\frac{1}{2}$ to $\frac{9}{16}$ inch	$\frac{3}{8}$ to $\frac{1}{2}$ inch
60 to 80 inches	$\frac{9}{16}$ to $\frac{11}{16}$ inch	$\frac{1}{2}$ to $\frac{9}{16}$ inch
80 to 100 inches	$\frac{11}{16}$ to $\frac{13}{16}$ inch	$\frac{9}{16}$ to $\frac{5}{8}$ inch
Above 100 inches	$\frac{13}{16}$ to $\frac{7}{8}$ inch	$\frac{5}{8}$ to $\frac{11}{16}$ inch

The general shape of the cylinder bottom and covers will be determined by the shape of the pistons used. The length of the cylinder must be such that a piston of the desired shape can travel the amount of the stroke, and have clearance at the top and bottom of the cylinder.

PISTONS.

The coned cast steel piston is the one most used, but sometimes the cast-iron box piston is used. The latter has the

shape shown in Fig. 6, and the usual proportions are as follows:

Depth of piston = $1.5d$ to $1.6d$, where d = diameter of piston rod.

Thickness of face of piston = $t = 0.0025 D \sqrt{P} + 0.33$ inch, where D = diameter of cylinder; $P = 0.5$ boiler pressure in high-pressure cylinder; $P = 0.25$ boiler pressure in medium-pressure cylinder; $P = 0.17$ boiler pressure in low-pressure cylinder.

Thickness near rim = $a = 0.9t$,

Thickness of ribs = $b = 0.9t$,

Thickness of boss around rod = $c = 1.75t$,

Thickness of junk ring flange = $e = 1.2t$,

Thickness of packing ring = $f = 0.75t$,

Breadth of packing ring = $g = 3t$,

Diameter of junk ring bolts = t

Diameter of junk ring bolts = $- + 0.25$ inch,

2

Pitch of junk ring bolts = 10 diameters.

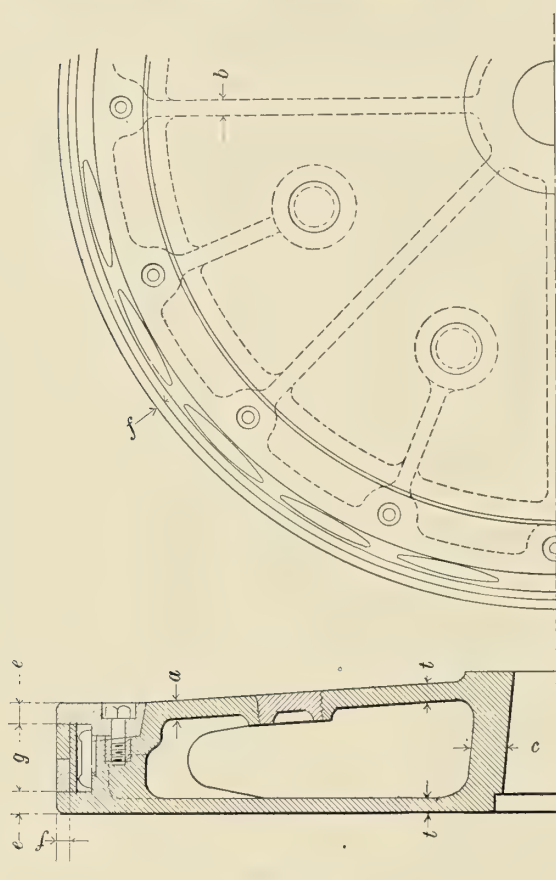


FIG. 6.

When cast steel pistons are used they are all made, when possible, of the same over-all depth, as shown in Fig. 7, the details of the rim being the same for all. The slope on the under side of the low-pressure piston varies from 1 in 3 to 1 in 6; the steeper this slope is made the longer the piston rod must be, and the greater the total height of the engine. The flange on the foot of the cylinder attaching it to the housings, or frame, must be sufficiently far below the under surface of the cylinder bottom to allow the nuts on the bolts in the flanges to be set up.

Figs. 8, 9 and 10 show some of the ways in which the cylinder bottom may be constructed to fit the pistons. The construction shown in Fig. 9 can be used when the low-pressure piston has a steep slope on the underside, but the construction shown in Fig. 8 is much stronger.

When cast steel pistons are used, the thickness at the rim

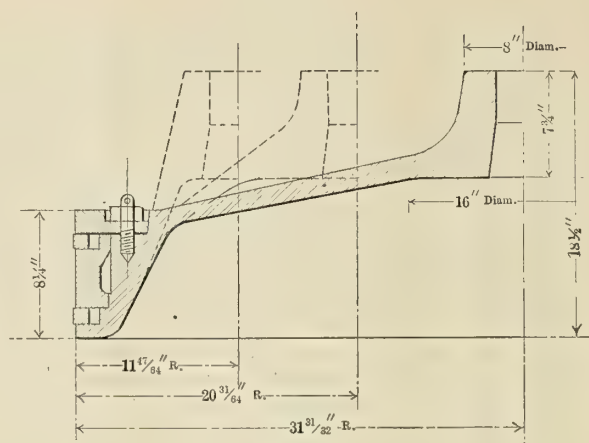


FIG. 7.

and at the center line of the boss can be obtained from the following formulæ:

$$\text{At the center, } t = \frac{D}{200} \sqrt{P + 0.25} \text{ inch.} \quad (16)$$

$$\text{At the rim, } t' = 0.5t, \quad (17)$$

where D = the diameter of the cylinder, and P may be taken as follows:

Ratio of Boiler Pressure for	Triple.	Quadruple.
For the high-pressure cylinder,	$P = 0.5$	$P = 0.45$
For the first medium-pressure cyl.,	$P = 0.25$	$P = 0.20$
For the second medium-pressure cyl.,		$P = 0.175$
For the low-pressure cylinder,	$P = 0.20$	$P = 0.10$

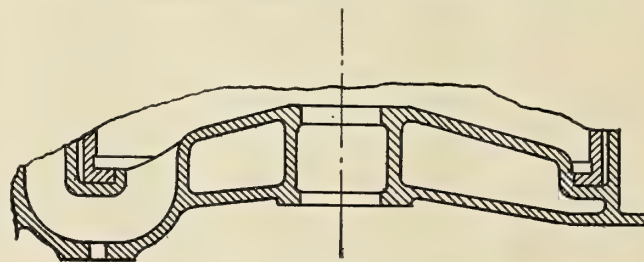


FIG. 8.

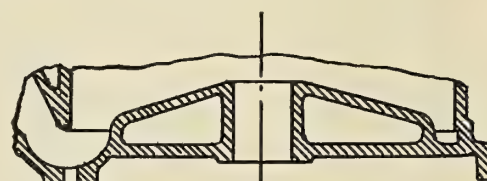


FIG. 9.

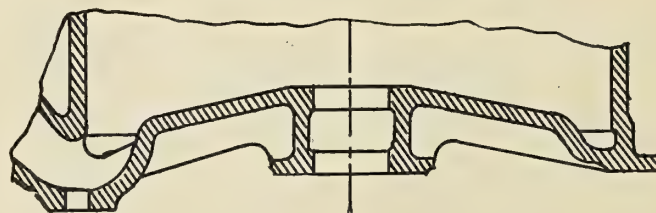


FIG. 10.

The construction of the pistons at the rim and at the boss is usually the same for all cylinders of the engine, except that, when the high-pressure cylinder is of small diameter, it is sometimes necessary to make a modification. Some of the constructions used for rims are shown in Figs. 11, 12, 13 and 14. The dimensions to be used in any case should be determined by the diameter of the low-pressure cylinder. The

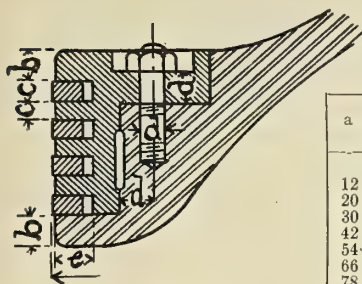


FIG. 11.

a	b	c	d	e	No. of Rings.
12	1 1/4	3/4	1	11/16	2 or 3
20	1 1/2	3/4	1	13/16	2 or 3
30	1 3/4	7/8	1 1/8	1 1/8	3
42	1 3/4	7/8	1 1/8	1 3/8	3 or 4
54	1 7/8	1	1 1/8	1 9/16	3 or 4
66	2	1	1 1/4	1 11/16	3 or 4
78	2	1 1/16	1 1/4	1 3/4	3 or 4

Dimensions in inches.

depth of the pistons at the boss should be from $1\frac{1}{2} D$ to $1\frac{1}{4} D$; $1\frac{1}{2} D$ for piston rods up to 6 inches in diameter, and ranging from $1\frac{1}{2} D$ to $1\frac{1}{4} D$ for rods from 6 inches to 11 inches in diameter. D is the diameter of the piston rod. The

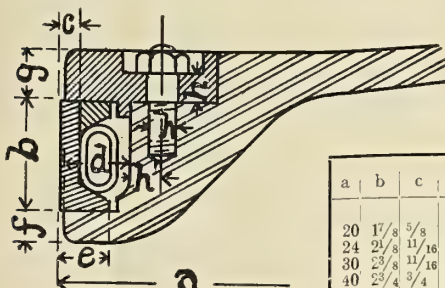


FIG. 12.

a	b	c	d	e	f	g	h
20	1 1/8	5/8	1 1/2	15/16	1 1/4	2 1/4	1
24	2 1/8	11/16	1 11/16	1 1/4	1 1/4	2 1/4	1
30	2 3/8	1 1/16	1 7/8	1 1/2	1 1/2	2 1/2	1 1/8
40	2 3/4	3/4	2 1/8	1 5/8	1 5/8	2 1/2	1 1/8
48	3	3/4	2 3/8	1 1/2	1 3/4	2 1/2	1 1/8
60	3 3/8	3/4	2 5/8	1 11/16	1 7/8	2 3/4	1 1/4
75	3 3/8	7/8	2 7/8	1 13/16	2	2 3/4	1 1/4
100	4	7/8	3 1/4	2	2 1/4	2 3/4	1 1/4

Dimensions in inches.

diameter of the boss should be sufficient to afford bearing for the nut on the end of the piston rod, the design of which will be taken up later. The underside of the piston at the center

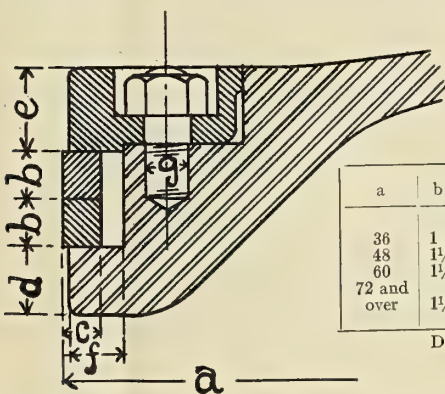


FIG. 13.

a	b	c	d	e	f	g
36	1	3/4	2	2 1/2	1 1/2	1 1/8
48	1 1/8	7/8	2 1/8	2 1/2	1 5/8	1 1/8
60	1 1/8	7/8	2 1/8	3	1 5/8	1 1/4
72 and over	1 1/4	1	2 1/4	3	1 3/4	1 1/4

Dimensions in inches.

is usually made horizontal for a distance from the center equal to

$$\frac{D}{2} + 1.5 \text{ inches,}$$

10

where D = the diameter of the low-pressure cylinder.

CYLINDER COVERS.

The shape of the underside of the cylinder cover is determined by the shape of the top of the piston. The distance a , Figs. 15 and 16, from the underside of the cover to the bottom of the flange, is determined by the height of the steam port entering the cylinder, and the thickness of metal above the port needed for the studs of the cylinder cover. The height of the port is generally from $4\frac{1}{2}$ to 7 inches, depending upon

the area necessary in order that the steam may enter with the desired speed.

The breadth of the joint between the cover and the barrel should be from $2.75 d$ to $3.25 d$, being usually $3 d$, where d is the diameter of the stud connecting the flanges. The studs

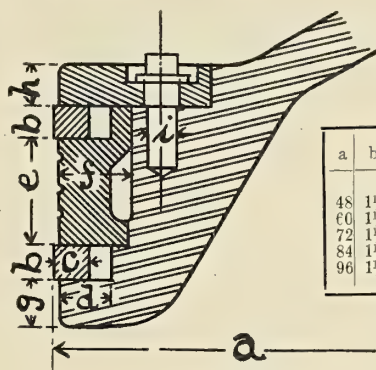


FIG. 14.

a	b	c	d	e	f	g	h	i
48	1 1/8	1	1 3/4	3	2 1/2	1 1/2	1 1/2	1 1/8
60	1 1/8	1	1 3/4	3	2 1/2	1 1/2	1 1/2	1 1/8
72	1 1/4	1 1/8	1 7/8	3 1/4	2 3/4	1 5/8	1 5/8	1 1/4
84	1 1/4	1 1/8	1 7/8	3 1/4	2 3/4	1 5/8	1 5/8	1 1/4
96	1 1/4	1 1/8	2	3 1/2	3	1 3/4	1 3/4	1 1/4

Dimensions in inches.

should be spaced far enough apart to allow a wrench to be used freely, and near enough together to make the joints steam tight. For the sake of convenience and simplicity, it is usual to calculate the studs which should be used for the high-pressure cover, and then to use the same size of stud upon the other cylinder covers, increasing the spacing as the steam pressure decreases.

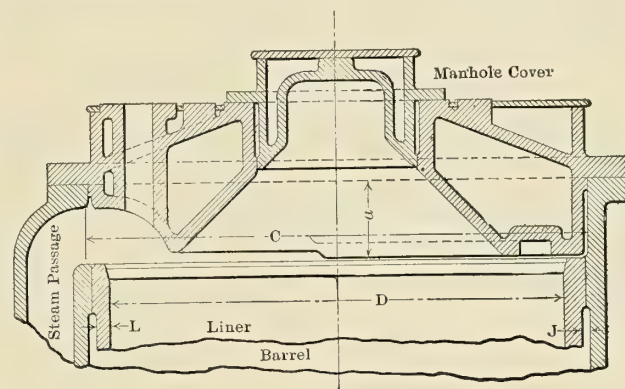


FIG. 15.

When a liner is used the inner surface of the barrel will have a diameter equal approximately to

$$C = D + 2L + 2J, \quad (18)$$

where D = the diameter of the cylinder,

L = the thickness of the liner,

J = the width of the jacket space, usually $\frac{3}{4}$ inch or 1 inch.

The load upon the cover will be the area of a circle of diameter C multiplied by the gage boiler pressure:

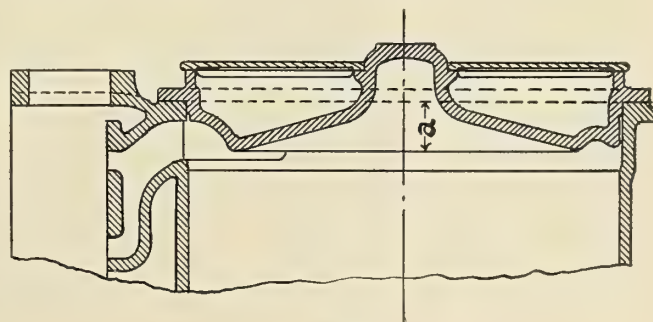


FIG. 16.

$$\text{Load on cover,} = \frac{\pi C^2}{4} \times (B.P.) \quad (19)$$

The cover studs should be spaced from $2.75 d$ to $3.25 d$ on the high-pressure cylinder, and the diameter of the pitch circle for these studs, upon the assumption that the width of joint is to be $3 d$, will be $C + 3 d$. The total load to be carried by the studs being known, a diameter of stud d , equal approximately to the thickness of the barrel, should be selected. Table IV. gives the load that the stud can carry, so that the total number necessary can be found.

$$\frac{(C + 3d)\pi}{\text{number of studs}} = \text{spacing of studs,}$$

which should be from $2.75 d$ to $3.25 d$. If the studs cannot be brought near enough together except by taking d less than $\frac{3}{4}$ inch, the diameter is taken as $\frac{3}{4}$ inch, and enough studs used to bring the spacing right. The spacing upon the medium-pressure cylinder cover should be from $4 d$ to $5 d$, and upon the low-pressure cover from $5 d$ to $6 d$. It is well to tabulate the results, as shown later in the calculations.

The nuts used on the cover studs are often thicker than the standard nut, in order that they may stand frequent handling. Covers for cylinders over 35 inches in diameter should have manholes in them, usually in the center of the cover. These manholes should be not less than 16 inches in diameter, and constructed as shown in Fig. 15. Manholes are frequently placed in the bottom of the cylinders, to give access to the underside of the piston without taking off the cover and removing the piston. These have to be placed where they will clear the upper end of the connecting rod.

CALCULATIONS FOR CYLINDERS AND PISTONS.

The use of the formulæ given will be illustrated by continuing the calculations for the engine whose cylinder diameters and cut-offs have been previously determined.

The assumptions and calculations made so far give us

I. H. P. = 3,000, P. S. = 850 feet per minute, B. P. = 185 pounds gage
Cylinder diameters and stroke, $23\frac{1}{2}$ ", 41", 64" $\times$ 42".

Cut-offs: H. P. = 0.675, M. P. = 0.60, L. P. = 0.65.

Cylinders.—Formula (15):

$$\begin{aligned} \text{H. P. cylinder liner and barrel, } t &= \frac{(185 + 25)23.5''}{6,000} + \frac{40''}{123.5} \\ &= 1.147'' (\text{Use } 1\frac{1}{8}'') \\ \text{M. P. cylinder liner and barrel, } t &= \frac{(100 + 25)41''}{6,000} + \frac{40''}{141} \\ &= 1.136'' (\text{Use } 1\frac{1}{8}'') \\ \text{L. P. cylinder liner and barrel, } t &= \frac{(50 + 25)64''}{6,000} + \frac{40''}{164} \\ &= 1.044'' (\text{Use } 1\frac{1}{16}'') \end{aligned}$$

Make the thickness of all cylinder barrels and liners $1\frac{1}{8}$ ".

Use double walls in covers and bottoms.

Thickness of cylinder bottom

$$= 0.9 \times 1.125'' = 1.01'' (\text{Use } 1'').$$

Distance between walls of cylinder bottom

$$= 5 \times 1.125'' = 5.625'', \text{ at least.}$$

Thickness of cylinder flanges

$$= 1.3 \times 1.125'' = 1.46'' (\text{Use } 1\frac{1}{2}'').$$

Width of cylinder flanges = $3 d$.

Thickness of metal in cylinder feet

$$= 1.125''.$$

Thickness of flanges on cylinder feet

$$= 1.5 \times 1.125'' = 1.69'' (\text{Use } 1\frac{3}{4}'').$$

Diameter of bolts for cylinder feet

$$= 1.4 \times 1.125'' = 1.58'' (\text{Use } 1\frac{5}{8}'').$$

Thickness of metal in cylinder cover

$$= 0.85 \times 1.125'' = 0.955'' (\text{Use } \frac{15}{16}'').$$

Maximum spacing of webs in bottom and cover:

$$\text{H. P., } \frac{1.125'' \times 100}{\sqrt{185}} = 8.3''. (\text{Use } 8\frac{1}{2}'')$$

$$\text{M. P., } \frac{1.125'' \times 100}{\sqrt{100}} = 11.25''.$$

$$\text{L. P., } \frac{1.125'' \times 100}{\sqrt{50}} = 15.9''. (\text{Use } 16'')$$

Thickness of metal in valve liners

$$= 1.125''.$$

Thickness of metal in ports and passages

$$= 0.9 \times 1.125'' = 1.01''. (\text{Use } 1'').$$

Piston clearances:

	H. P.	M. P.	L. P.
Top.....	$\frac{3}{8}''$	$\frac{3}{8}''$	$\frac{9}{16}''$
Bottom.....	$\frac{1}{2}''$	$\frac{1}{2}''$	$\frac{11}{16}''$

Pistons.—Formula (16):

$$\frac{23.5''}{200} \times \sqrt{100} + 0.25'' = 1.43'', \text{ high-pressure.}$$

$$\frac{41''}{200} \times \sqrt{50} + 0.25'' = 1.70'', \text{ medium-pressure.}$$

$$\frac{64''}{200} \times \sqrt{40} + 0.25'' = 2.03'', \text{ low-pressure.}$$

Formula (17): $0.5 \times 1.43'' = 0.715''$, high-pressure.

$$0.5 \times 1.70'' = 0.85'', \text{ medium-pressure.}$$

$$0.5 \times 2.03'' = 1.015'', \text{ low-pressure.}$$

Depth of piston at boss = $1.5 \times 5.25'' = 7.87''$. (Use $7\frac{3}{4}''$.)

Diameter of cylindrical nut on top of piston rod = $4'' \times 1.75 = 7''$
(See below.) Top of boss to be 8 inches diameter.

Radius of horizontal portion of the underside of piston

$$\frac{64}{10} + 1.5'' = 7.9''. (\text{Use } 8'').$$

The diameter of the body of the pistons will be:

$$\text{High-pressure, } 23\frac{1}{2}'' - \frac{1}{32}'' = 23\frac{15}{32}''.$$

$$\text{Medium-pressure, } 41'' - \frac{1}{32}'' = 40\frac{31}{32}''.$$

$$\text{Low-pressure, } 64'' - \frac{1}{16}'' = 63\frac{15}{16}''.$$

The details of the piston are shown in Figure 7.

Bolting for Cylinder Covers.—Formula (18):

$$C = 23.5'' + 2 \times 1.125'' + 2 \times 0.75'' = 27.25''.$$

$$\text{Load on cover} = \frac{\pi(27.25'')^2}{4} \times 185 = 108,000 \text{ pounds.}$$

Try $1\frac{1}{8}$ -inch studs of 70,000 pounds ultimate strength, which (see Table IV.) can carry 3,115 pounds each.

$$\frac{108,000}{3,115} = 35. (\text{Approximately.})$$

$$27.25'' + 3 \times 1.125'' = 30.625'';$$

$$\text{Circumference of pitch circle} = 30.625 \times \pi = 96''; \quad \frac{96''}{35} = 2.75'';$$

$$\frac{2.75''}{1.125''} = 2.44 \text{ diameters; (Too close).}$$

1.125''

Try $1\frac{1}{4}$ -inch studs. Working load = 4,110 pounds.

$$\frac{108,000}{4,110} = 26. (\text{Approximately.})$$

$$27.25'' + 3 \times 1.25'' = 31''; \quad 31'' \times \pi = 97.3'';$$

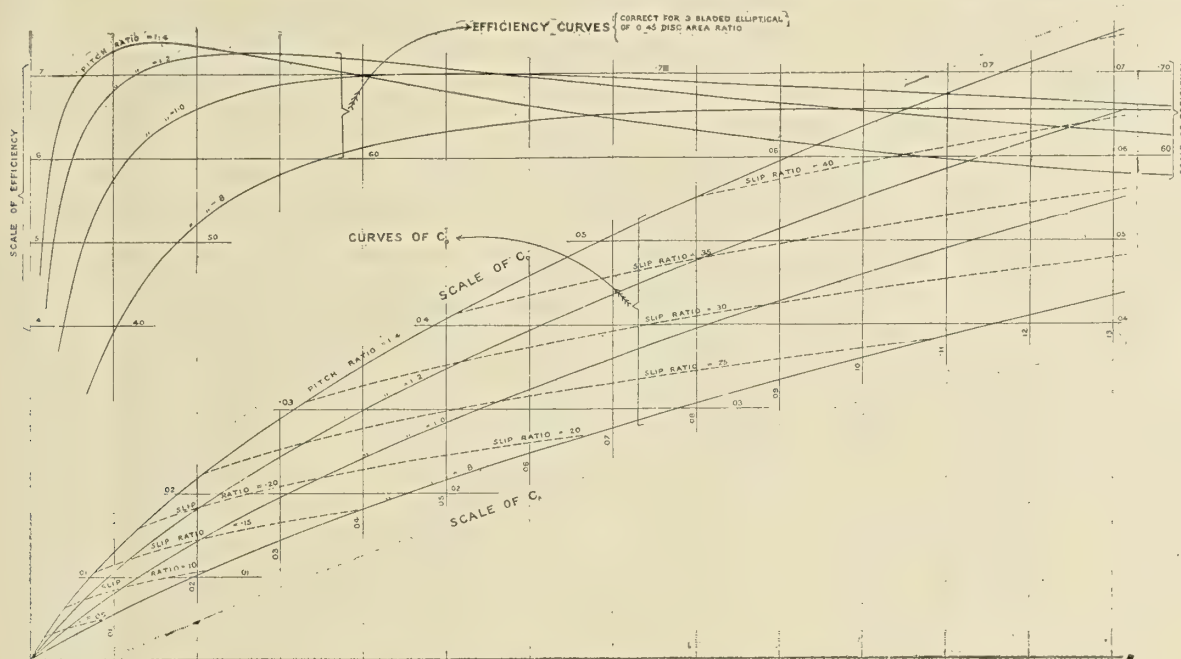
$$\frac{97.3''}{26} = 3.75''; \quad \frac{3.75''}{1.25''} = 3 \text{ diameters; (Use } 1\frac{1}{4}\text{-inch studs.)}$$

Cylinder Cover Studs.	Diameter of Stud.	Pitch of Circle of Studs.	Number of Studs.	Spacing of Studs, Diameters.
Cylinder High.....	23.5"	1.25"	31"	26
Medium....	41"	1.25"	48½"	29
Low.....	64"	1.25"	71½"	33

(To be Continued.)

course, far beyond the experimental data; this being done to help guide the directions of the curves at the confines of the data. In thus extending the compass of the curves, as well as in fairing them throughout, regard was had, so far as the experiment data left room for any question, to the theoretically calculated curves in Figs. 10 and 11, on which I comment later.

Design.—The "x y" curve, with its companion efficiency curves, serves perfectly for analysis of steam trials, where it is needed only to calculate thrust horsepower and efficiency, for known revolutions per minute and speed, with known propeller dimensions. But, for design, where it is generally

FIG. 7.—EFFICIENCY CURVES OF C_0 , BASED ON SCALE OF C_0 .

RESULTS OF FURTHER MODEL SCREW PROPELLER EXPERIMENTS.*

BY R. E. FROUDE, F. R. S.

Efficiency.—We have next to consider the efficiency reduction. The variation of efficiency with pitch ratio makes it impossible to bring all the results under one single efficiency curve, as was done for the 1884 experiments. It was therefore necessary to obtain a series of efficiency curves for a convenient series of round-number pitch ratios, such as appear in Fig. 6, plotted to the base "x" of the "x y" curve. This diagram, as well as that in Fig. 7, to avoid confusion, shows curves for an abridged series of pitch ratios. These are correct for the three-blade elliptical type of screws, and for a standard disk-area ratio of 0.45. To correct them for another type or disk-area ratio, or both, a uniform efficiency reduction must be made, of .020 for three-blade wide tip, or .0125 for four-blade elliptical, also one appropriate to the disk-area ratio and pitch ratio in question, as indicated by the ordinates of the efficiency correction curves, to base disk-area ratio, in Fig. 5.

Efficiency for a pitch ratio intermediate between those for which the series of efficiency curves are shown has to be obtained by interpolation, but in nearly all cases of practical occurrence the successive curves are so near together that this can readily be done by eye.

The series of efficiency curves for the series of round-number pitch ratios, already referred to, were obtained from those for the four pitch ratios of actual experiment, which appear in Fig. 8, by means of cross curves, such as are shown in Fig. 9. In the former diagram, the curves are carried to unity slip ratio, and in the latter to zero of pitch ratio; in both cases, of

desired to calculate propeller dimensions suitable to given horsepower, revolutions and speed, a difficulty arises from the fact that diameter, an unknown quantity, enters into both x and y. So, for purposes of design, there were devised the curves in Fig. 7, for which the abscissa "CA" is determined solely by the known horsepower, revolutions and speed; while the corresponding ordinate " C_0 " determines the unknown diameter in terms of the same known quantities. The expressions for these two values are

$$CA = \frac{R^2 H}{B V^5} \left[\frac{p + 21}{p^3} \cdot x^2 y \right] \quad (7)$$

$$C_0 = \frac{H}{B D^2 V^3} \left[\frac{p + 21}{p} \cdot y \right] \quad (8)$$

Unlike the "x y" curve, this curve differs with pitch ratio; a series of curves is therefore shown, for the same series of pitch ratios as the series of efficiency curves accompanying the "x y" curve, which series of efficiency curves is also reproduced on the CA base to accompany the curves of C_0 .

The x y curves can be used for design, though with indifferent convenience, as follows: If we choose a value for p, a series of chosen values of x, with the corresponding ones of y, determine corresponding ones of D from V and R by equation (5), and thence ones of B from V and H by equation (6). Thus we can get curves of D to base B (or propeller area if preferred) for a series of values of p. The main objection is that for chosen p there is a very small range of choice of x without exceeding the practical range of B.

Unfortunately, it is never practicable to treat pitch ratio as

* Read before the Institution of Naval Architects, April 10, 1908.

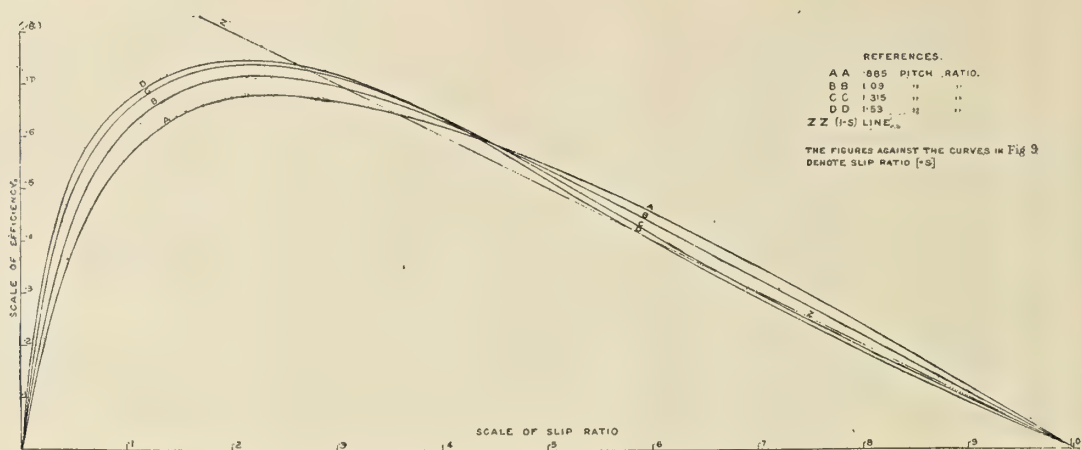


FIG. 8.—EXPERIMENTAL EFFICIENCY CURVES, BASED ON SLIP RATIO.

an unknown quantity. Our usual practice in using these data is to obtain diameter and efficiency for two or more of the pitch ratio values for which curves are given, each for two or more values of disk-area ratio, and plot the results on a base of total blade area. In this way the diameter and efficiency for any intermediate pitch ratio is indicated. The disk-area ratio being reckoned for total area of outline without boss allowance, in computing the total blade area for ship propeller from the diameter and disk-area ratio, a discount must be made, to allow for portion of area covered by boss. In our general practice at Haslar this discount is 20 percent.

Nominal Pitch.—It may be recollected that in the system of analysis and reduction just described, the pitch P , corresponding both to the pitch ratio p and the slip ratio S , is throughout taken as equal to the travel per revolution for zero thrust; in other words, the revolutions of zero thrust are taken as the zero of the slip-ratio scale. Comparison of the figures which we have calculated from these data with the realization in actual ships have led to the conclusion that the pitch of pitch ratio figures used or obtained in these calculations, which we may term "analysis pitch," should be taken as 1.02 times the nominal (or driving-face) pitch for ship.

For present purposes, I prefer this result of net experience to any estimate based on the relations between "analysis" pitch and constructional pitch in the model screws, partly because I question the adequate nicety of the pitch molding of the model propellers for such a purpose, partly because of the differences of condition between ship propellers in use, and model propellers under experiment, some of which are referred to above.

THEORETICAL EFFICIENCY CURVES.

The curves shown in Figs. 10 and 11 have been computed on three different bases. No. 1 of these is the solution of Mr. William Froude's paper of 1878,† obtained from the rotary and thrust components of the estimated edgewise and normal resistance of an elementary plane, mounted on a revolving radius, obliquely to the plane of rotation. The other two are the same in principle (slightly different mathematics), but the resistance forces of the supposed plane or blade are obtained from the experiments made at Haslar some years ago on planes and blades moving obliquely at various angles. In No. 2 the plane was oval and thin, with edges symmetrically sharpened; in No. 3 the *blade* (as it must more properly be termed) was rectangular, and flat all over the front face, but rounded on the back like a screw blade, the mid-thickness being about one-seventeenth of the width. In both the length was twice the width, and the length was transverse to the line of motion.

The planes or blades were tried at various angles of obliquity from the path, and the components of force, R in line of path and L transverse to it, measured. I am indebted to Mr. Arnulph Mallock for the following charmingly simple solution of the efficiency of such plane or blade mounted on a revolving radius and treated as an elementary screw propeller, as in Mr. William Froude's solution. In figure C let AB be the path of plane (A towards B), AC the plane of rotation, CB the line of axis, and V the speed of advance (C towards B), then we shall have

(1) In respect to force L ,

† Transactions Institution of Naval Architects, Vol. XIX., page 47.

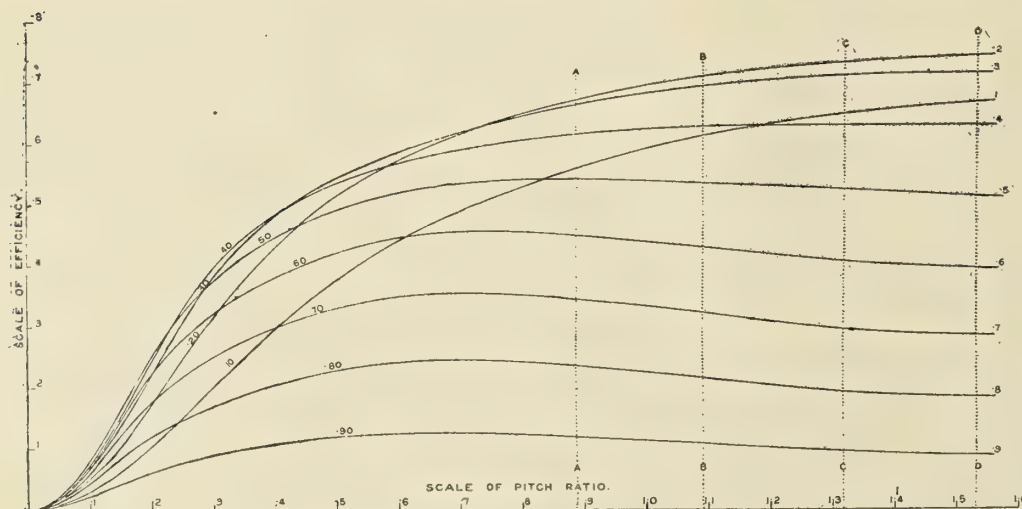


FIG. 9.—EXPERIMENTAL EFFICIENCY CURVES, BASED ON PITCH RATIO.

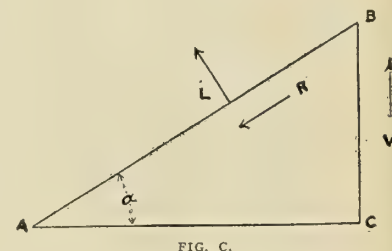


FIG. C.

$$U_L, \text{ i.e., Useful power } = V L \cos \alpha,$$

$$E_L, \text{ i.e., Expended power } = V L \cos \alpha.$$

(2) In respect to force R ,

$$U_R, \text{ i.e., Useful power } = - V R \sin \alpha,$$

$$E_R, \text{ i.e., Expended power } = V \frac{\cos \alpha}{\sin \alpha} R \cos \alpha = V R \frac{\cos^2 \alpha}{\sin \alpha}$$

slip, in this respect according with the results of the screw experiments, as shown in Fig. 8. The curves in Fig. 8 terminate at zero of slip ratio, while those of Fig. 10 do not, because the latter are calculated for pitch of surface, but the former for pitch equal to travel per revolution at zero thrust, therefore necessitating zero efficiency at zero of slip ratio. This circumstance turns on a rather important theoretical point, as follows:

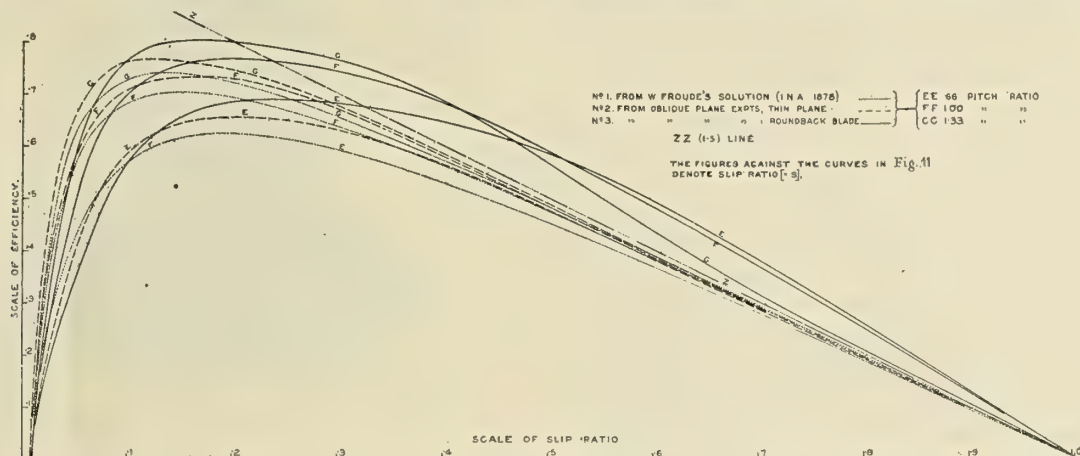


FIG. 10.—THEORETICAL EFFICIENCY CURVES, BASED ON SLIP RATIO.

whence

$$\text{Efficiency} = \frac{U_L + U_R}{E_L + E_R} = \frac{1 - \frac{R}{L} \tan \alpha}{1 - \frac{R}{L} \cot \alpha}$$

or, writing

$$\frac{R}{L} = \tan y$$

$$\text{Efficiency} = \frac{\tan \alpha}{\tan (\alpha + y)}$$

In the ideal case of a screw with no edgewise resistance, where, in other words, the turning moment must be simply the virtual-velocity component of the thrust, the sole waste would be slip, and the efficiency would be always $= (1 - S)$. This expression, indicated by the ordinates of the straight line ZZ in Figs. 8 and 10, may therefore be regarded as the theoretical limit of efficiency, from which an actual screw must fall short in virtue of the edgewise resistance element. But it will be seen that the actual screw curves in Fig. 8 begin to trespass *outside* this line before even 30 percent slip is reached. The comparison between the theoretical curves for bases Nos. 2 and 3 proves clearly that this feature is incidental to the roundness of back of the blades, which must obviously operate to increase the effective pitch [and so falsify the $(1 - S)$ line]; and this not by a constant amount, since the increase at no slip has already been taken account of in the mode of assessing the

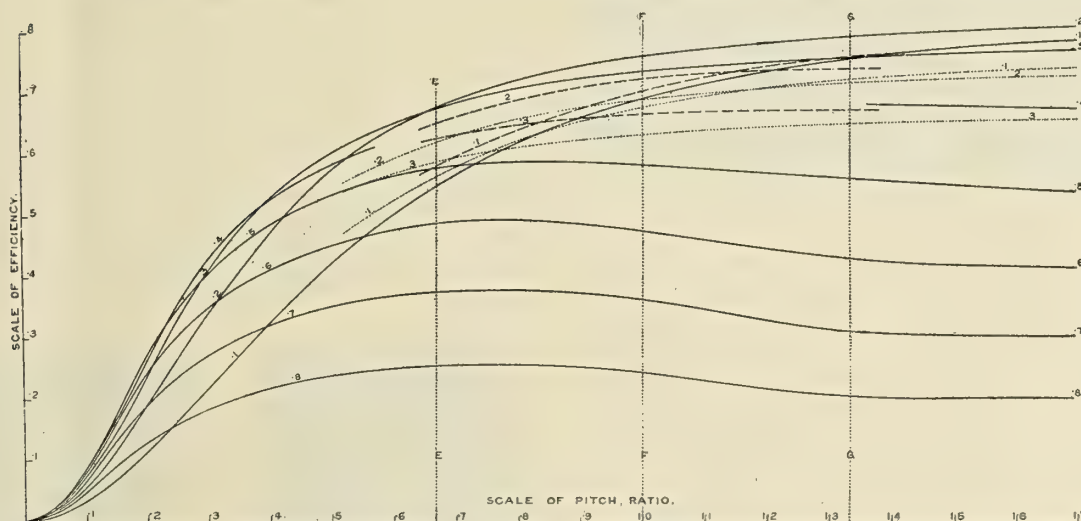


FIG. 11.—THEORETICAL EFFICIENCY CURVES, BASED ON PITCH RATIO.

This was the solution used in computing the curves for bases Nos. 2 and 3.

It will be seen that in Fig. 10, Nos. 1 and 2 follow each other pretty closely, as might be expected, but that No. 3 differs from the two others in a striking way in the regions of higher

analysis pitch, but by an amount which increases markedly as slip ratio increases.

In reference to Fig. 11, it should be noted that the ordinates of the curves for basis No. 3 have been calculated for the following pitch ratio values: .05, .10, .20, .33, .50, .66, 1.0, 1.33,

1.66, 2.0, 2.4. The finish tangential to the base line at the zero pitch ratio end is consonant with theory, because, in the limiting case, for given revolutions per minute, turning moment and work expended are constant, whereas for given slip ratio, both thrust and speed vary as pitch, and useful work consequently as pitch squared. The pitch ratio figures assigned to the theoretical curves in Fig. 10, and to which they are plotted in Fig. 11, are taken as two-thirds of the values proper to the actual supposed path of the plane or blade; on the supposition that the mean diameter of an ordinary screw may be taken as about two-thirds of the reputed diameter.

THE HEATING AND VENTILATING OF SHIPS.

BY SYDNEY F. WALKER, M. I. E. E.

NON-LUMINOUS HEATING APPARATUS WITH LOOSE POWDER.

There is another form of electric heating apparatus, which has been developed in Germany, principally, in which a loose powder is employed, the necessary resistance being obtained partly by means of the substance of which the powder is composed and partly by the fact that the substance is in a powder, or in loose grains. A loose powder, or loose contact between any two conductors, across which an electric current has to pass, always offers a considerable resistance over and above that due to its own sectional area, length, etc. This is the cause of the heating of badly designed switches. If the contact portions of a switch do not make good contact with each other, heat is always liberated at the surfaces, and sometimes arcs are formed with the attendant enormous heat.

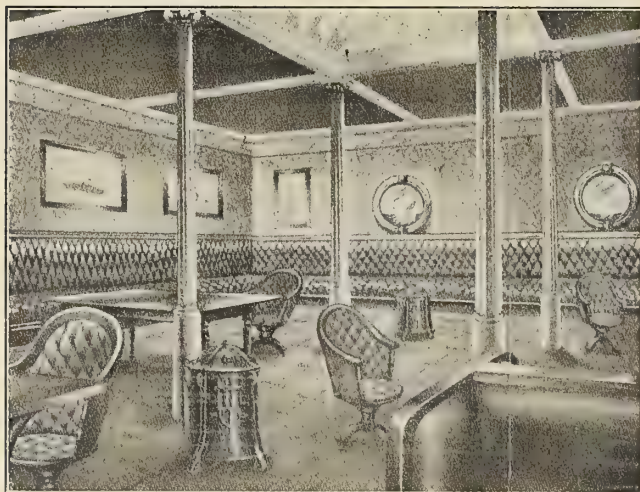


FIG. 35.—SALOON ELECTRICALLY HEATED BY HAWKES' STOVES.

One form of this heating appliance is known as "Kryptol." It is a granular mass of very inoxidizable substances, carbon, graphite, carborundum and silicious matters. These substances are ground together and then pressed into blocks, and afterwards made into grains of a uniform size. The grains for different types of apparatus vary in size from a sand to the size of grains of wheat, with varying amounts of graphite and carborundum, according to the particular applications for which they are required. The substance is claimed to stand temperatures up to 3,000° F.; and, on the other hand, it is claimed that temperatures as low as 50° F. can be obtained.

The powder or grain is filled into cartridges, as shown in Fig. 36. These cartridges consist of tubes of special glass, in which the resistance material is held, the ends of the tubes being hermetically sealed with metallic capsules, which form the connections to the powder. The cartridges are heated with an electric current before they are finally closed by the cap-

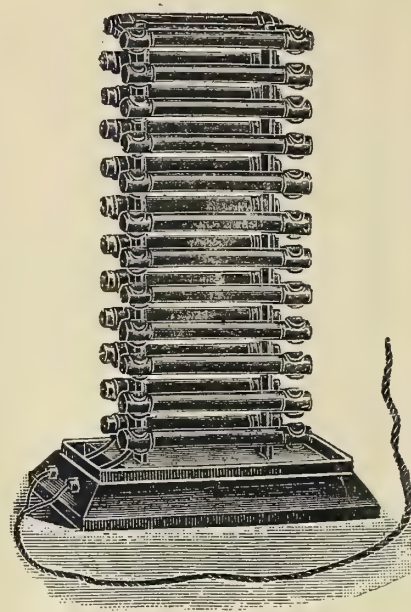


FIG. 36.—KRYPTOL CARTRIDGES BUILT INTO FRAME.

sules, in order to eliminate grains of un-uniform size, and also to get rid of the moisture. One of the troubles met with in working out this form of apparatus, after the capsules had been fixed, was the generation of steam within the cartridge when the current was allowed to pass through, the steam bursting the glass-containing tubes. To meet this difficulty, any moisture that may be present is driven off by the heat of an electric current, the moisture forming steam, and the heating being kept up until this has all disappeared, and the whole mass is thoroughly dry, and until dry air is present between the grains of the substance. The cartridges are built into various forms, and are arranged as radiators, or convectors, whichever term may be preferred, some of which are shown in Figs. 37 and 38.

Kryptol is also used, in certain cases, in what are practically open fireplaces. The grains are loosely heaped in a vessel of fireproof clay, the current being led to the mass by conductors projecting into them. For other purposes also the Kryptol grains are spread loosely on a plate that it is desired to heat, or in an annular space surrounding an object to be heated, etc.

The action of the substance is as follows: When the current is first switched on, small arcs are sometimes formed between the individual grains, this leading to the very rapid development of heat. But in the cartridge tubes, providing that they are properly prepared, it is claimed that the formation of arcs has been practically suppressed. In either case, whether arcs are formed or not, the substance settles down usually to a dull red heat, which may be increased up to the high temperature named, if sufficient current is passed through



FIG. 37.

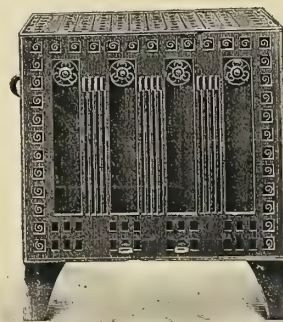


FIG. 38.

KRYPTOL CABIN HEATERS.

it for a sufficient time. Where the substance is used loose, practically in air, the formation of the arcs mentioned leads to the burning away of the substance itself by the formation of carbonic oxide and carbonic acid, just as in an ordinary furnace or in an arc lamp. It is stated, however, that the powder can remain, with the current passing through it, for several hours before it need be renewed.

Some tests that have been made upon a stove intended for heating rooms and containing twenty cartridges inside a cover of expanded metal are interesting. They are shown by the curve in Fig. 39. In the figure the ordinates are temperatures in degrees Centigrade, and the abscissæ represent time in minutes. The stove was used to heat up a room, whose initial temperature was 10°C . (50°F .), the outside temperature being 2°C . (35.6°F .). The current employed was 9 amperes, with a pressure of 120 volts—a little over one kilowatt, or Board of Trade unit.

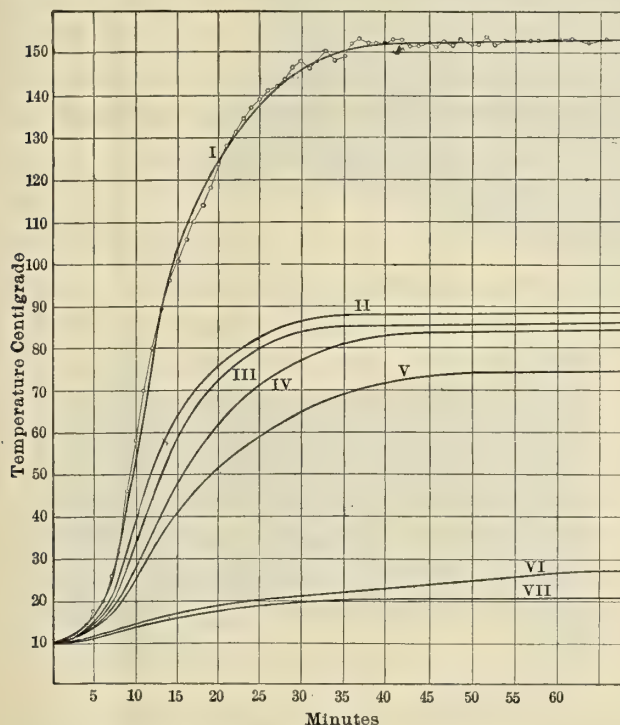


FIG. 39.—TEMPERATURE CURVES OF CARTRIDGE STOVE.

In the figure, line I. shows the variation of the temperature of the air between the two upper cartridges, with the cover of the stove removed. The cartridges were in two vertical rows of ten each, and the temperatures given would be between the two upper ones, just inside the top of the stove. It will be seen that the temperature rises in 10 minutes from 10°C . (50°F .) to 60°C . (140°F .). In 15 minutes it has risen to 100°C . (212°F .); in 20 minutes to 124°C . (255°F .). After this the rise is more gradual, reaching 150°C . (302°F .) in 35 minutes, and 152°C . (305.6°F .), at which it remains constant to the end of the test, which occupied an hour. Curve II. shows the temperature on the center of the cover, which was presumably replaced. It will be noticed what a great difference there is between the temperature of the cover and that of the air inside of the apparatus between the cartridges. The rise of temperature is still very equal, and it reaches 65°C . (149°F .) in 15 minutes, but it reaches 88°C . (190°F .) only in 35 minutes, and does not rise any higher to the end of the test. Curve III. is the temperature of the air of the room 20 millimeters ($\frac{3}{4}$ inch) above the top of the cover. It will be noticed that the temperature follows the same course as that of the cover itself, but is about 4°C ., say 7°F ., less. Curve IV. is the temperature of the top of the frame carrying the

cartridges, which, it will be seen, follows the course of curves II. and III. fairly closely, with a certain difference between them. Curve V. is the temperature of another portion of the cover, not subject to side currents of air. It does not present much interest. Curve VI., which is the most interesting one of the whole, is the temperature of the room one meter ($39\frac{3}{8}$ inches) above the cover; and curve VII. is the average temperature of the air in the room. It will be seen that the temperature of the air, one meter above the cover, and the average temperature of the room, are very nearly alike, that a short distance above the cover being slightly higher than the average temperature, and being about 7°C . (12.6°F .) above it at the end of the test. Both curves, however, rise very gradually. It takes 25 minutes to increase the temperature 10°C . (18°F .) one meter above the stove, and 30 minutes for

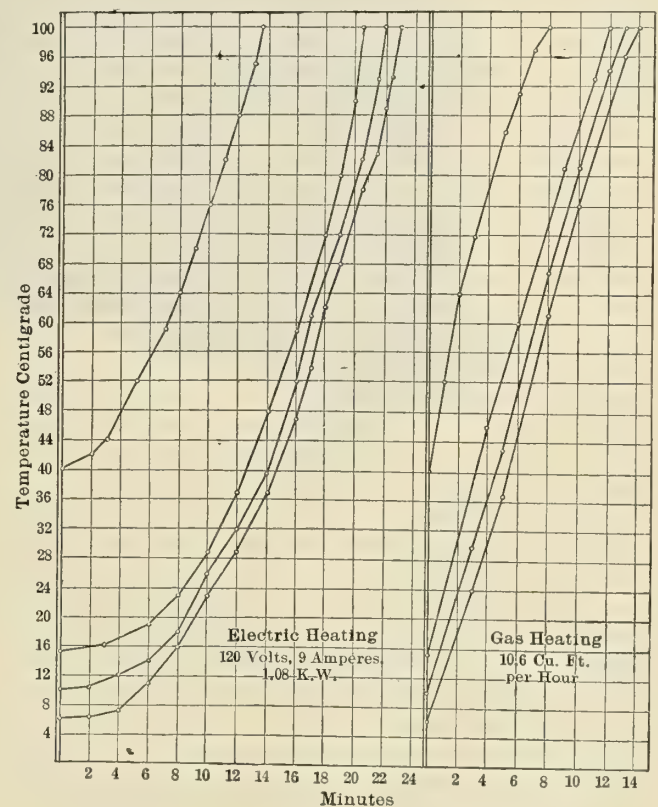


FIG. 40.—HEATING TO BOILING POINT OF ONE LITER OF WATER.

the average temperature of the room to reach the same figure. The temperature of the air, one meter above the stove, rises very gradually, it will be seen, to about 27°C ., while the average temperature of the room rises to only about 21°C . (70°F .).

There is another instructive series of curves given of tests with a Kryptol stove, shown in Fig. 40. There are several curves, those on the left of the figure giving the rise of temperature in the time shown, with the Kryptol apparatus, and those on the right the rise of temperature in the time shown, with gas. The gas employed appears not to have been by any means the most efficient for heating. It was an open gas flame, which is certainly not designed for heating. As will be seen, the electrical apparatus takes 10 minutes to reach a temperature of 28°C . in the best of the three curves shown, and over 20 minutes to reach 100°C ., while gas reaches 30°C . in the worst of the curves shown in 4 minutes, and 100°C . in 14 minutes.

The above curves are taken from an article in the *German Export Zeitschrift*, dealing with the subject. The article also gives some other interesting information, which there is hardly

space to reproduce here. Some other curves are given, which show that the current required rises to a maximum, and then falls to a "working" current. This is the common experience with a great many forms of heating apparatus. If the air of a room is required to be heated up quickly, a considerable amount of heat has to be supplied to the apparatus for a short time, and then it may be reduced, the air then keeping its temperature with a smaller expenditure.

SPIRAL COIL HEATERS.

A type of electric heater made by the Consolidated Car Heating Company, New York, and fitted for marine use, is on the McElroy spiral coil construction, in which the resistance coils are perfectly supported at every point, rendering vibration impossible. The spindle supporting the coil consists of a $\frac{3}{8}$ -inch square wrought iron rod, on which are strung porcelain tubes, so designed and fitted that a helical groove

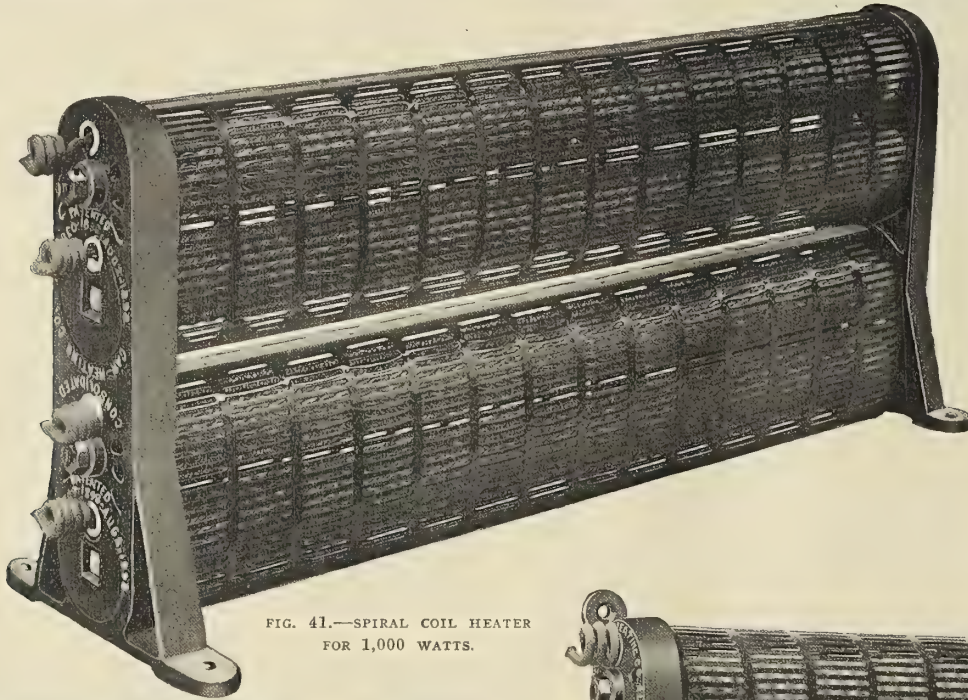


FIG. 41.—SPIRAL COIL HEATER FOR 1,000 WATTS.

The Kryptol cartridges described are made in the following sizes: $5\frac{1}{2}$, $7\frac{1}{4}$, 10, $12\frac{3}{4}$ and 20 inches long, by 0.6 and 0.8 inch diameter respectively. The $12\frac{3}{4}$ -inch cartridge takes 0.3 ampere with a pressure of 100 volts, and the cartridge is stated to receive with that current and pressure an increased temperature of 100 degrees. These figures are for the cartridge when exposed. When inclosed, the conductivity of the mixture rises with the temperature, and the cartridge will take 0.4 ampere with 110 volts.

It will be understood, as explained in connection with hot water and steam heating, that the above remarks apply to heating, without having any regard to the question of ventilation. As will be explained, heating and ventilation are now usually considered together, the ventilating air current being employed for heating and cooling purposes. In many cases, however, no attention whatever is paid to ventilation, and this is particularly the case with electrical heating appliances.

It will be understood that any heating appliance may be fixed in any room, passage, alleyway, etc., and will give off heat, exactly in the proportion described, but the heat given off may or may not be useful heat. In the case of corridors, one very frequently sees here a heating appliance, which is practically useless, because there is an air current constantly passing over it, and constantly carrying off the heat that is liberated, without doing any useful work. The same remark would apply to a room that is very subject to drafts. The heating appliance would do very little good. On the other hand, if a heating appliance is placed in a room, say in the middle of a saloon, and is not exposed to drafts, it will heat up the air of the saloon, by radiation and convection, in a certain time, varying with the conditions, but the heating will be hardly under the control of the engineer in the same manner as it is when the appliances are so arranged, as will be described later, as to utilize the warmed air currents.

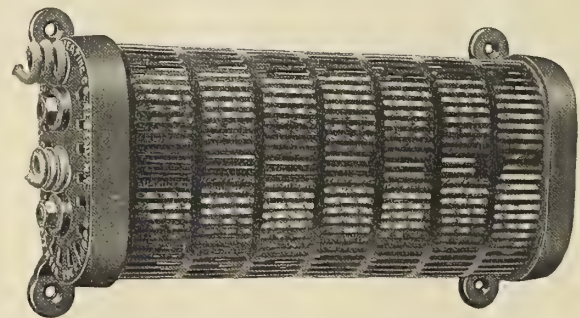


FIG. 41A.—SMALL SPIRAL COIL HEATER FOR STATEROOM.

extends from end to end. The iron wire for the resistance coil is wound in a close spiral spring, and insulated copper leading wires are attached to both ends by twisted and soldered joints. This coil is wound between the ridges on the porcelain spindle, under suitable tension, and the leading wires are passed through eccentric bushings at the ends and firmly fastened to the exterior part of the circuit. This construction gives the greatest possible length of wire in the given space, and so disposes every portion of the large surface presented that a large quantity of air comes freely into contact with it, and passes out in a steady stream at such temperature as may be designed.

Two views of this type of heater are given in Figs. 41 and 41A. The former is designed for the use of 1,000 watts (1 kilowatt) of current, measures 27 inches in length, $15\frac{3}{8}$ in height and $3\frac{9}{16}$ in thickness. The "spread" for bolt holes is 7 inches, and the heater, which contains four coils, is finished in black japan. The smaller heater shown has only two coils, and measures $4\frac{1}{8}$ inches in thickness, with a spread of $6\frac{1}{2}$ inches. The length is $15\frac{3}{8}$ inches, while the oval of the case measures $5\frac{3}{4}$ by $3\frac{3}{8}$ inches. The case is of heavy, perforated sheet steel.

LOW TEMPERATURE AIR HEATER, TUBULAR TYPE.

The latest in air heater design is the so-called "low temperature air heater." The specifications of the United States

battleship *Louisiana* called for electric heaters which should have an operating temperature equivalent to that of steam piping. As a result, the General Electric Company's engineers designed the tubular type of air heater shown in the accompanying illustration, and furnished it to the *Louisiana*. This particular type of heater consists of three or more tubular heating elements inclosed by the metal "chimney" tubes which are shown. Each tube dissipates 250 watts. The principle of this design is the combination of the large radiating surface with a low watt surface density and the chimney effect of the tube.

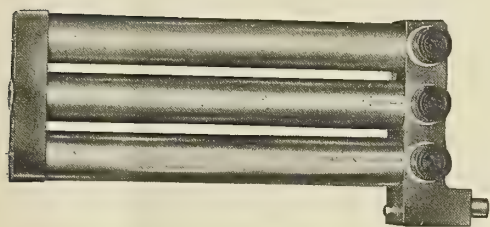


FIG. 42.—TUBULAR TYPE, LOW-TEMPERATURE AIR HEATER.

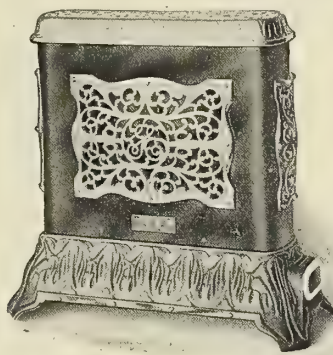


FIG. 43.—AIR-HEATER, CARTRIDGE UNIT TYPE.

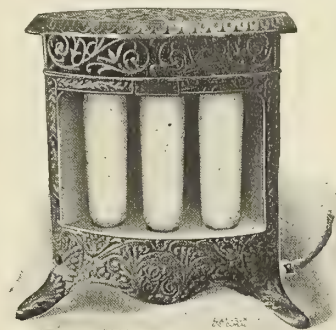


FIG. 44.—GENERAL ELECTRIC LUMINOUS RADIATOR.

It is manifest that, while all electric air heaters may be said to give 100 percent efficiency, the *practical* efficiency, which is judged by the uniform and effective distribution of the heat in the room, can be obtained only by passing a relatively large volume of air over the heating surfaces, and raising it only a few degrees above the temperature of the room. The oftener the total volume of air in the room passes over the surface of the heating source, and the less temperature difference between the outlet and inlet, the more efficient is the heating system.

Three distinct forms of heating elements are used by the General Electric Company. The cartridge unit consists of a thin tape of special resistance metal, wound edgewise, insulated with a fireproof cement and then inserted in a mica-lined brass tube capped with a cement plug through which the leading-in wires are brought. The quartz enamel unit is made up of a resistance wire wound in a coil of small diameter, which is then coiled into the form of a flat spiral, with mica insulating strips between its convolutions, and held against a layer of quartz grains imbedded in enamel on the bottom of the heater. Both of the foregoing heating units are practically infusible and indestructible, but can be readily replaced if damaged by accident. Great care has been taken in the design of the heating devices to insure the most efficient application of the heat, and at the same time to give proper radiating surface, so that nearly all the apparatus may be left in circuit indefinitely without fear of burn-out. (See Fig. 33.)

The third form of heating element is the tubular resistance, which is used in the tubular air heater already described. This resistance, while designed only for comparatively low temperatures, is one of the cheapest and best forms for air heaters up to a maximum of 600 or 700 degrees F., or with a density of 2 or 2½ watts per square inch. It was first developed by the General Electric Company for rheostat work, and particularly the heavy service of the railway rheostat. It consists of a tube of asbestos wound on a mandrel, the tube supporting a single layer of resistance wire closely wound, but with turns not touching. The tube is then impregnated with a fire-proof insulating compound, which gives the asbestos considerable stiffness and forms a protecting coat over the resistance wire.

REGULATING THE HEAT DELIVERED BY ELECTRIC HEATING APPARATUS.

The favorite method is similar to that described in connection with the glow lamp radiator. The heating elements are arranged inside the apparatus, in such a manner that either each element individually, or groups of elements, can be switched in and out at will. The usual arrangement is, for heating appliances, the heating elements are connected in parallel between what are practically two bus-bars, connected to the supply service. There is a main switch to disconnect the whole appliance, and there are subsidiary switches to



FIG. 45.

PROMETHEUS REGULATOR DETAILS.



FIG. 46.

connect and disconnect either individual elements, or groups of elements, from the bus-bars.

The British Prometheus Company has another system of regulating for some of their apparatus, which is something on the lines of the regulator of the tramway service. There is a sleeve of approximately square section, as shown in Fig. 45, with conductors on the insides of the four faces, connected to the heating elements. The corresponding fitting (Fig. 46) consists of a solid piece of insulating material of square section, carrying conductors on its faces, the conductors being connected to flexible cords, to which the regulator is attached. It is arranged that the conductors on the male portion, when making connection with certain conductors on the female portion, allow full, three-quarters, one-half or one-quarter of the current strength to pass as may be desired, the arrangement being made by connecting the different elements in the heating appliance in different order. Thus, for full heat, all the elements will be connected in parallel. For half heat, two sets will be connected in parallel, afterwards being connected in series in each parallel, and so on, for the other heats.

Other methods of varying the heat include that shown in Fig. 47, which is adopted by Messrs. Isenthal, of London, which is somewhat similar, though different in form, to that of the Prometheus Company. The heating apparatus has three projecting pins as shown, and the connecting pipe from the supply service has three plug holes. When the three plug holes are on the three pins, the full current is passing, and the full heat is liberated. When the two plug holes on the left engage with the two pins on the right, the medium current is passing, and

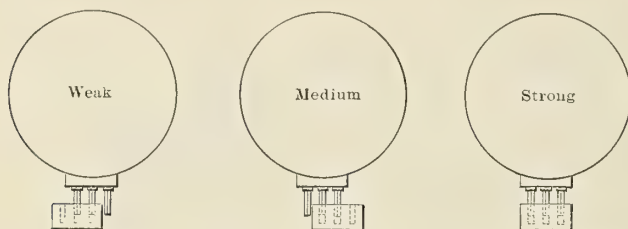


FIG. 47.—ISENTHAL METHOD OF REGULATING WITHOUT SWITCHES.

when the two plug holes on the right engage with the two pins on the left, a weak current is passing. The strength of the currents under this arrangement are as one, two and three. The three-hole plug is wired with twin wire, one of the twins being connected to the center plug hole, and the other to the two outside plug holes.

The Prometheus Company, of New York, has a somewhat similar arrangement for regulating the heat in certain cases. There are three pins on the heating apparatus, and there are three terminals on porcelain holders, connected to three conductors of a flexible cord. The three terminals on the flexible cord are colored, one red and the others black. By different arrangements of the terminals by engaging the red terminal and the black terminals with different pins, different heats are provided.

THE QUANTITY OF HEAT LIBERATED IN ELECTRICAL HEATING APPARATUS.

Referring to the formula, H is given in watts, when E is given in volts, C in amperes, and R in ohms; these being, as marine engineers know, the units of electrical power, pressure, current and resistance. The watt is the unit of power or the rate of doing work, and it will be familiar to engineers from the fact that 746 watts equal one horsepower. Work is done at the rate of one watt, when a current of one ampere passes with a pressure of one volt, or the equivalent. Thus, in the ordinary 16-candlepower incandescent lamp, working with a pressure of 100 volts, and taking a current of 0.6 ampere, the electrical energy expended in each lamp equals $100 \times 0.6 = 60$ watts.

Coming to the heat question, each watt liberates 0.0568 British thermal unit per minute, or 3.41 British thermal units per hour. These figures are derived from the figures already given, showing that the heat unit equals 17.58 watts. It is claimed, by makers of electrical heating apparatus, that the whole of the electrical energy delivered to the apparatus, whether it be in the form of the lamps that have been described, or any one of the resistance materials mentioned, is converted into heat; and therefore, where an electrical heating apparatus is employed to heat a room, the whole of the electrical energy is applied in heating the air and objects in the room. The writer mentions the claim, and so far scientists appear to have assumed that the principles upon which it is based are correct.

It is assumed by scientists that every form of energy, when transformed from the state in which it is at any moment, becomes heat sooner or later—that heat is the final form of all energy, and that the heat balance sheet is the final court of

appeal upon all matters in which any form of energy is concerned. It appears to the writer that it is quite possible that other forms of energy may be liberated, when electricity is converted into something else. The question whether this does take place, or not, has not yet been examined in any way by scientists, and therefore the above statement is given with all due reserve, and the calculations which follow will be understood to be subject to that reservation. If all the electricity delivered to the heating apparatus becomes heat, the calculations are correct. In any case, it appears to the writer that any difference there may be would come within the margin which every practical engineer allows himself for possible sources of error.

The electric lamps described above, which are employed in luminous radiators, absorb, as mentioned, 250 watts each, and that would mean that $250 \times 3.41 = 852\frac{1}{2}$ British thermal units are liberated by each lamp per hour. As each British thermal unit raises the temperature of 55 cubic feet of air 1 degree F., each lamp will raise the temperature of 47,000 cubic feet of air 1 degree F. in one hour, or, say, approximately, 4,700 cubic feet 10 degrees F. in one hour, two and four lamps raising the temperature of proportional quantities of air to the same degree.

Leaving out for the moment the question of air currents and ventilation, which will be dealt with further on, it is a simple calculation to find the number of lamps required to raise the temperature of a room of a given cubical content through a given number of degrees. The temperature to which the air has to be raised varies, of course, with the climate and the seasons, but taking 40 degrees F., the figure worked to in the calculations which follow, as the increase of temperature required, this would be provided for in a room having a cubical content of 1,175 cubic feet, by one of the lamps mentioned, in one hour, on the supposition that all of the electricity is converted into heat, and that no heat passes out of the room during the time.

The above remarks apply equally to non-luminous radiators, which are made to take various quantities of electricity. Apparatus is made absorbing from 500 up to 4,000 watts, when taking their full current, and liberating from 1,700 to 13,600 heat units per hour. They are usually made to regulate the current at one-quarter, one-half and three-quarters of the full heating capacity, the heat units liberated being then from 425 to 3,400 with one-quarter, and the other figures in proportion.

(To be Continued.)

American Shipbuilding in 1908.

Returns covering the fiscal year 1908 show that 1,506 vessels of 588,627 gross tons were built. The largest previous annual output was in the year 1855, when 2,024 vessels, of 583,450 tons, were built. The steel vessels built in the year just ended numbered 142, of 417,167 gross tons (average, 2,938 tons), compared with 131, of 366,517 gross tons (average, 2,798 tons), built in 1907. The record for steel construction was broken in both these years. Of these totals for steel vessels the Great Lakes accounted for 75 of 304,379 tons in 1908, and 47 of 238,713 in 1907.

Of the 142 steel vessels built in 1908, 85 exceeded 1,000 gross tons each. Of these 55 were built on the Great Lakes, the largest being the *William M. Mills*, of 7,962 tons, and 30 were built on the seaboard, the largest being the *Columbian*, of 8,579 tons, built at San Francisco for trade to Hawaii. Four ocean sailing vessels exceeding 1,000 tons each were built during the year, the largest being the *Edward J. Lawrence*, of 3,350 gross tons. The tonnage built in 1908 was entirely for domestic transportation, no vessels exclusively for foreign trade having been launched in the United States.

THE COMBINATION SYSTEM OF RECIPROCATING ENGINES AND STEAM TURBINES.*

BY HON. C. A. PARSONS, C. B., F. R. S., D. SC., M. A., AND R. J. WALKER.

In the early years of steam turbine design and development it became apparent that the turbine engine was capable of economically dealing with ratios of expansion far beyond the reach of any reciprocating engine, whose limitations in this respect had been experimentally determined by many investigations.

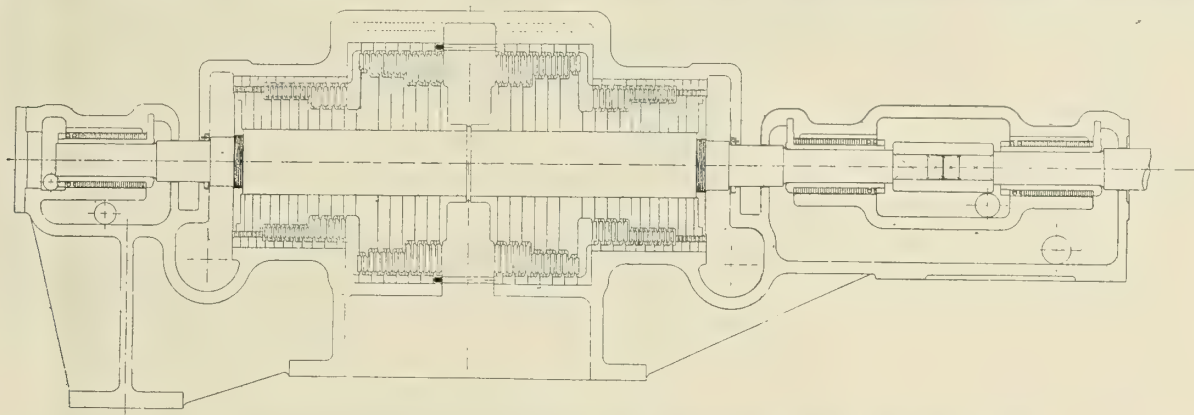


FIG. 1.—FIFTY-HORSEPOWER LOW-PRESSURE TURBINE FOR ATMOSPHERIC PRESSURE, DESIGNED IN 1889.

In 1889 the first condensing turbine, of about 100 horsepower, was designed for an expansion ratio of 100 by volume, the expansion being effected in two turbines of the double-parallel flow type, the low-pressure turbine (Fig. 1) taking steam from the exhaust of the high-pressure at atmospheric pressure, and expanding it down to 1 pound absolute. The striking feature presented by this design was the very high estimated efficiency of this low-pressure portion. A separate low-pressure turbine was not, however, actually constructed until some years later.

In 1894 a patent was taken out for the "combination" of a reciprocating engine with a steam turbine, whose object was "to increase the power obtainable by the expansion of the steam beyond the limits possible with reciprocating engines."

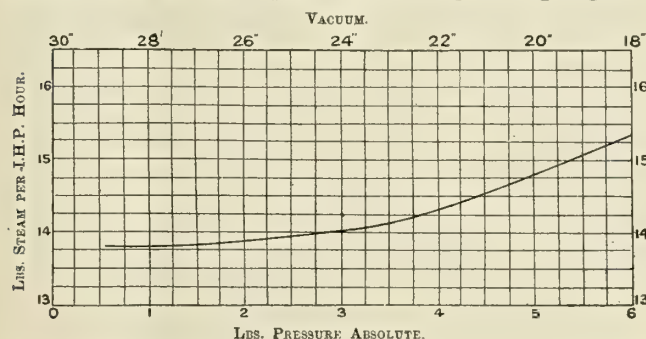


FIG. 2.—EFFECT OF VACUUM ON STEAM CONSUMPTION OF 200-HORSEPOWER TRIPLE EXPANSION RECIPROCATING ENGINE.

The previous treatment of the steam is, of course, immaterial, provided that its condition of pressure and wetness on reaching the engine are known.

The first instance of a separate turbine worked from the exhaust of other turbines was in the *Turbinia's* machinery in 1897—the pressure at entry of her low-pressure turbine was about 9 pounds absolute, and the exhaust 1 pound absolute. The slip ratio of her three shafts showed that the low-pressure turbine developed about one-third of the total horsepower obtained from the steam at 160 pounds pressure, agreeing closely with calculations.

In the year 1902 the combination of reciprocating engines

exhausting into turbines was first put to a practical test in His Majesty's destroyer *Velox*. In this vessel two small reciprocating engines were fitted for cruising purposes, of such power that, in combination with the main turbines, they would give an economical consumption at speeds of 11 to 13 knots, the usual cruising speeds at the time the *Velox* was built.

The arrangement of machinery consisted of one main high-pressure and one low-pressure turbine on each side of the vessel, each driving a separate shaft, or four shafts in all. The two small reciprocating engines were coupled at the forward end of each of the low-pressure turbines. For speeds up to

about 13 knots, steam was admitted to the two reciprocating engines, and expanded down to about atmospheric pressure; it then passed through the high-pressure, and thence through the low-pressure turbines to the condenser. This combination gave excellent results at these cruising speeds. For speeds above 13 knots, however, the reciprocating engines had to be cut out and steam admitted to the turbines alone. With the advance of naval efficiency, the cruising speeds of war vessels

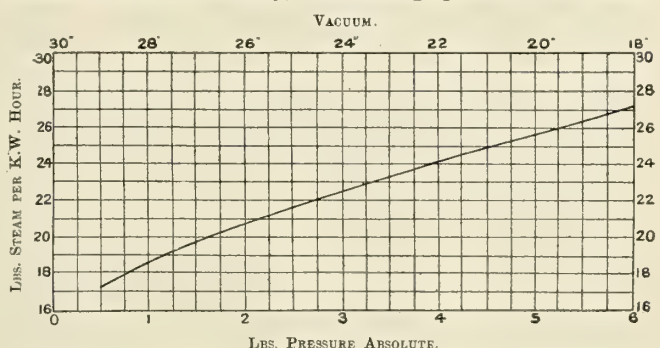


FIG. 3.—EFFECT OF VACUUM ON STEAM CONSUMPTION OF 1,000-KILOWATT TURBO-GENERATOR.

have been increased, and in vessels subsequent to the *Velox* additional high-pressure turbines have been fitted, an arrangement which permits of good economy over a wide range of cruising speeds.

It may be said that perhaps the most important field for the combined system of machinery as applied to marine propulsion is for those installations where the designed full speed of the vessel falls below the range suitable for an all-turbine arrangement, the reciprocating engine working in the region of pressure-drop where the conditions are best suited for it, and the turbine utilizing that portion of the expansion diagram which the reciprocating engine is not able to utilize efficiently. It is generally well known that an all-turbine arrangement has not been advocated by us for ships where the designed speed falls below 15 or 16 knots, excepting in some special cases, such as yachts; and for vessels of moderate or slow speed the combination system of machinery appears to be eminently suitable.

* Read before the Institution of Naval Architects, April 9, 1908.

In a good quadruple reciprocating engine, the steam is expanded down to the pressure of release, about 10 pounds absolute, and gains in economy as the vacuum is increased up to about 25 or 26 inches, whereas, in a turbine, it is possible to deal economically with very low-pressure steam, and to expand this low-pressure steam to a low absolute pressure corresponding to the highest vacuum obtainable in turbine practice.

Figs. 2 and 3 show the effect of vacuum upon steam consumption as the result of tests carried out on a reciprocating engine and steam turbine, respectively, from which it will be noted that, while the curve for the reciprocating engine gives

of between 8 pounds and 16 pounds absolute, or even at a slightly higher pressure, if necessary, to meet the conditions required. From an estimate of the theoretical efficiency under the various conditions of pressure as set forth in the following table, it would appear, apart from any practical considerations, that there is nothing to choose between an initial pressure at the turbine of 7 pounds and 15 pounds absolute, any pressure within this limit appearing to give a most economical result. The assumption is for 200 pounds absolute steam pressure at the reciprocating engine, and 28 inches vacuum at the condenser:

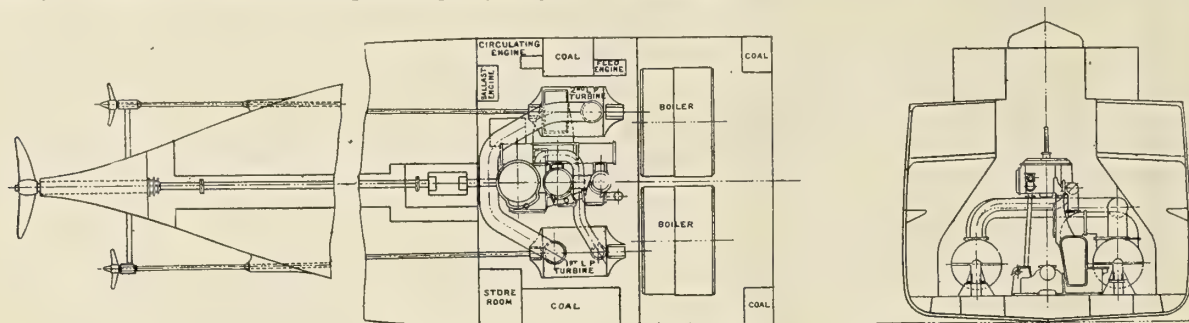


FIG. 4.—ARRANGEMENT OF COMBINATION OF TRIPLE-EXPANSION ENGINE WITH TWO LOW-PRESSURE TURBINES IN SERIES.

the minimum consumption at between 25 and 26-inch vacuum, the curve for the turbine continues to fall as the vacuum increases.

In a certain quadruple expansion reciprocating engine exhausting to condenser direct, the maximum energy realizable, from 200 pounds to 26 degrees vacuum, with point of release at 10 pounds, is 256 British thermal units. The additional area which the reciprocating engine cannot efficiently utilize, but which can be used in a turbine, is 73 British thermal units. In the combination of triple expansion reciprocating engine exhausting to turbine and thence to condenser, the maximum energy realizable in the engine from 200 pounds to 8 pounds absolute, with point of release at 13 pounds, is 219 British thermal units. The energy available for the turbine, from 7 pounds to 28-inch vacuum, receiving wet steam from the reciprocating engine, is 100 units. The total energy of the combination is 319 British thermal units. Theoretically, the total energy of combination is 24½ percent greater than that

Initial Pressure, Turbine.	Reciprocating Engine, Back Pressure.	THEORETICAL B. T. U. PER POUND OF STEAM.		
		Engine.	Turbine.	Total.
15	16	178	142	320
12½	13½	189	131	320
7	8	218	100	318

In the case of a vessel which runs on service continually at or about her designed full speed, an initial pressure of about 7 pounds absolute at the turbine appears most suitable. In a vessel which does part of her running at the designed power, and part at a considerably reduced power, it is desirable to design the turbines so that the initial pressure would not fall below 7 pounds absolute when running under the lower conditions of power.

It might be of interest at this stage to consider the disposition of the turbines in combination with reciprocating

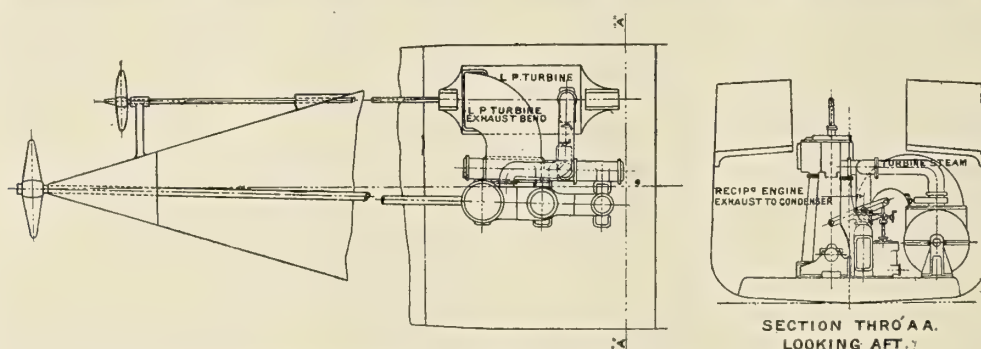


FIG. 5.—ARRANGEMENT OF COMBINATION OF ENGINE WITH ONE LOW-PRESSURE TURBINE.

of the quadruple engine. It is estimated that a large portion of this additional energy can be realized by the combined system in the shape of increased power to drive the vessel, or, on the other hand, increased economy. The theoretical figures are computed on the basis of adiabatic expansion throughout.

In a combination system, the most suitable initial pressure for the turbine, or the dividing line between the reciprocating engine and the turbine, will greatly depend upon the conditions of service of the particular vessel taken. The reciprocating engine, or engines, could be designed to exhaust at a pressure

engines on board ship. The arrangement of the turbine, or turbines, depends greatly upon whether the vessel is to be fitted with single or twin-screw reciprocating engines. With a single reciprocating engine, one turbine, two turbines in "series," or two turbines in "parallel" could be fitted, each turbine driving a separate shaft, in addition to the reciprocator shaft. With twin-screw reciprocating engines, an arrangement of one turbine in the center of the vessel, two turbines in "parallel," or two turbines in "series," could be adopted. The arrangement which seems to commend itself generally to ship-owners and builders, where twin-screw reciprocating engines

are fitted, is the arrangement with the turbine on the center shaft.

In 1901 and the two or three following years, alternative schemes were prepared from time to time. Fig. 4 shows an arrangement which was prepared in 1901 of a single reciprocating engine in combination with two low-pressure turbines in "series." The indicated horsepower of this proposal was 1,500, speed $11\frac{1}{2}$ knots, and loaded displacement 5,300 tons. Fig. 5 shows another arrangement of one reciprocating engine, and a turbine on one side of the vessel. The indicated horse-

estimated amount of saving in consumption, so that the total indicated horsepower of the combination did not exceed that required with twin quadruple engines. This would considerably reduce the total weight of machinery, and also the bunker capacity for a given distance. This saving in the weight of the machinery and in the bunkers would enable the vessel to carry an equivalent addition in deadweight cargo. Then, again, if we take the indicated horsepower at 8,300 for the combination, and assume that quadruple engines and boilers were required to give an equivalent power, the extra total weight of

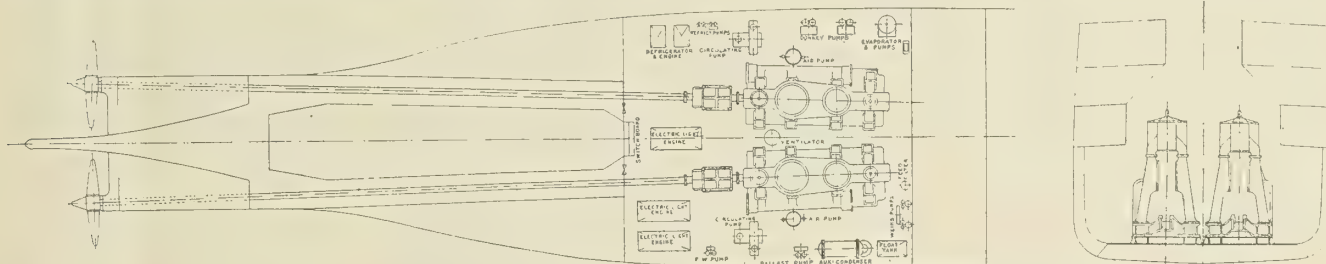


FIG. 6.—GENERAL ARRANGEMENT OF TWIN-SCREW QUADRUPLE-EXPANSION ENGINES OF 7,300 HORSEPOWER.

power of this vessel was 1,200 total, and the speed 10 knots.

The turbines in each proposal were designed for about 25 percent of the power, and the reciprocating engines for the remainder, the turbines taking the steam from the reciprocating engines at 7 pounds absolute pressure. It was estimated that the combination, as applied to cargo vessels, would be about 15 to 20 percent more economical than the ordinary triple expansion engine usually fitted to this class of vessel.

Owing to the rapid development of the turbine industry for high-speed work, and the attention which was, in consequence, paid to this branch of the business generally, the development of the combination system fell more into the background than its merits and the wide scope of its application would seem to have deserved. About two years ago, at the suggestion of Sir William White, designs were prepared of a combination system as applied to the intermediate type of liner of moderately large power and speed, and since that time numerous designs have been prepared for various types of vessels of speeds ranging from 13 to 16 knots.

By the courtesy of Swan, Hunter & Wigham Richardson, who, for a considerable period, have taken much interest in

	A.	B.	C.
Cylinders of reciprocating engines	25, 36½, 51½, 75	27, 42, 66	26, 39, 46, 46
Stroke of pistons	55	48	42
Revolutions of reciprocating engines	84	85	100
Piston speed of reciprocating engines	770	680	700
Boiler pressure	213 pounds	213 pounds	213 pounds
Estimated pressure at high-pressure receiver	200 pounds	200 pounds	200 pounds
Initial pressure, absolute, turbines	7 pounds	7 pounds	7 pounds
Vacuum at condenser	26 inches	28 inches	28 inches
Steam consumption, per hour, main engines only	95,000 pounds	95,000 pounds	95,000 pounds
Indicated horsepower, reciprocating engines	7,300	6,300	6,300
Estimated equivalent indicated horsepower of turbines	2,000	2,000	2,000
Total indicated horsepower	7,300	8,300	8,300
Percent increase in power	13.7	13.7	13.7
Estimated speed of ship	15.5 knots	16.2 knots	16.2 knots
Steam consumption, per total indicated horsepower per hour, main engines	13 pounds	11.45 pounds	11.45 pounds
Steaming weight of engines and boilers (reciprocating engines)	1,430 tons	1,455 tons	1,455 tons
Weight of turbine installation	65 tons	70 tons	70 tons
Total steaming weight	1,560 tons	1,495 tons	1,525 tons
Pounds per total horsepower	479	403	412
Revolutions of turbines	480	480	320

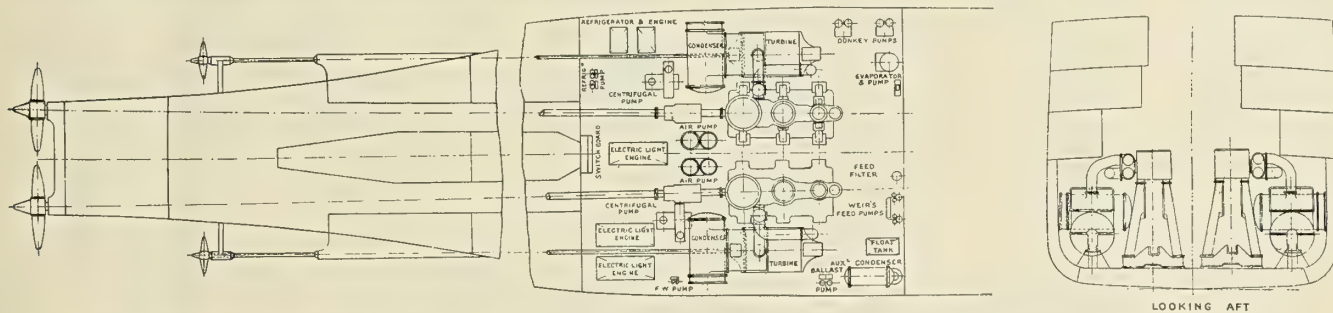


FIG. 7.—ARRANGEMENT OF COMBINATION OF TWIN-SCREW ENGINES AND TWO LOW-PRESSURE TURBINES IN PARALLEL; 8,300 HORSEPOWER.

this combination of machinery, the figures in the second table are given of the comparative sizes of engines, power, etc., of the "combination" as compared with twin-quadruple engines, for a proposed steamer of 490 feet in length, 13,600 tons displacement, 7,200 indicated horsepower, and $15\frac{1}{2}$ knots speed.

In the combination proposals set forth in columns B and C in the table, it may be mentioned that in this particular inquiry the shipowners wished to have the advantage of the additional power and increase in speed of the vessel on the same coal consumption as for the twin-quadruple engines. In some instances an increase in speed might not be desired, in which case the boilers and engines could be reduced in size by the

machinery would be, roughly, 160 tons, in addition to an increase of about 12 percent in coal consumption for the same power.

Fig. 6 shows an arrangement of twin-quadruple engines corresponding to particulars given in column A (twin-quadruple expansion reciprocating engines) of the table. Fig. 7 shows an arrangement of twin triple expansion engines, working in conjunction with two low-pressure turbines, as set forth in column B (twin three-cylinder triple expansion engines, with two low-pressure turbines in parallel), the reciprocating engine on each side exhausting into a turbine. By this arrangement an independent set of engines is obtained on each side of the

vessel. Fig. 8 shows an arrangement of four-crank triple expansion engines and one low-pressure turbine, as set forth in column C (twin four-cylinder triple expansion engines with one low-pressure turbine).

In arrangements shown in Figs. 7 and 8, for maneuvering in and out of port, suitable arrangements are made for changing the flow of steam of the low-pressure cylinder exhaust of the reciprocating engine from the turbine to the condenser. This can be done in two or three ways. One method is to have an ordinary change valve of the piston type, or ordinary double-beat spring loaded valve actuated by links, connected to the weigh shaft of the main engine, which would automatically change the flow of steam to the condenser when the engine was reversed. With this arrangement, when going ahead on one side of the ship, the steam from the reciprocating engine would flow through the turbine; but there does not appear to

watts, and representing a total brake horsepower of about 17,000. In most cases the exhaust steam is supplied to the low-pressure turbine at 15 pounds absolute pressure, and a vacuum of 28 inches, and under such conditions about an equal amount of power can be obtained from the turbine as from the non-condensing reciprocating engine; thereby doubling the power of the plant without any further consumption of fuel. From several tests made with these exhaust turbines on land, a consumption of about 34 pounds per kilowatt-hour can be obtained in a 500-kilowatt machine, with an initial pressure of 15 or 16 pounds absolute, to 28-inch vacuum.

In regard to marine installations, the combination is being fitted to a large vessel at present under construction at the works of Harland & Wolff, for the Montreal trade of the Dominion Line. The arrangement of machinery in this vessel is substantially as described in column C of the table previously

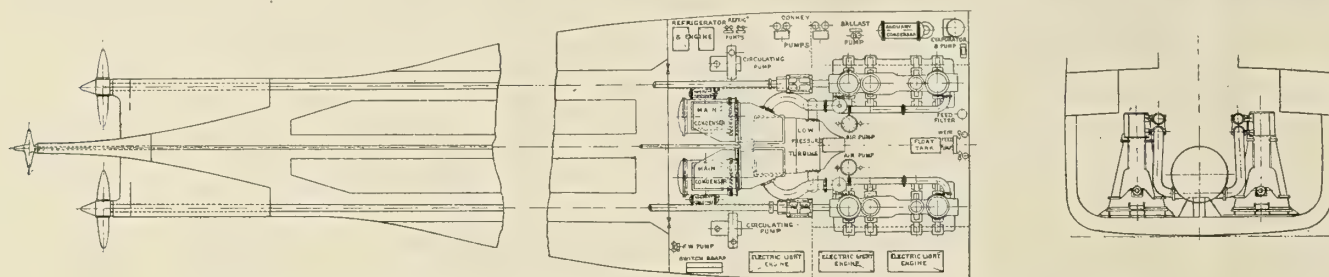


FIG. 8.—ARRANGEMENT OF TWIN-SCREW ENGINES AND ONE LOW-PRESSURE TURBINE; TOTAL HORSEPOWER, 8,300.

be any objection to this, even if we consider the twin-screw reciprocating arrangement with a single turbine on the center shaft. It might be rather an advantage, than otherwise, to allow the steam from the engine going ahead to pass through the turbine, as the center propeller revolving would accelerate the feed of water on the rudder, and augment the turning power of the vessel. Figs. 9 and 10 show such arrangements.

Another method would be to work these valves independently of the main engines, actuated by a hydraulic engine, or by an

referred to, viz.: twin four-crank triple expansion engines, exhausting into one low-pressure turbine driving the center shaft.

William Denny & Brothers are also at present building a vessel for the New Zealand Shipping Company, which is being fitted with the combination system of twin triple expansion engines and one low-pressure turbine. This vessel is an exact repeat of two other vessels which Messrs. Denny have built for the same owners, except as regards type of engines.

In addition to the above, the Turbinia Works, in conjunction with Alexander Stephen & Sons, of Glasgow, are fitting the combination system to the yacht *Emerald*. In this vessel, which was one of the first to be fitted with an all-turbine arrangement, it is intended to make some modifications to the

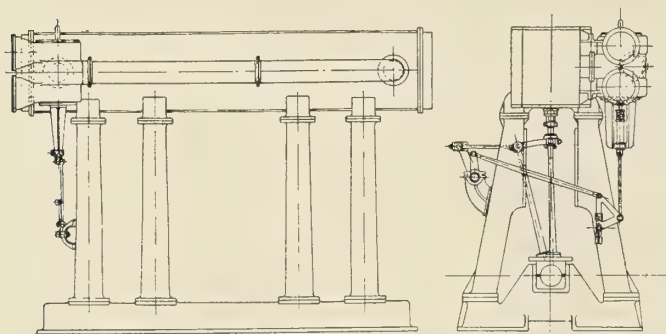


FIG. 9.—ARRANGEMENT OF PISTON TYPE CHANGE VALVES BETWEEN RECIPROCATING ENGINE, TURBINE AND CONDENSER, IN COMBINATION SYSTEM WORKED FROM MAIN ENGINE.

ordinary steam-driven reversing engine. With this arrangement the low-pressure turbine would be cut out altogether, and the reciprocating engine would exhaust to the condenser, whether going ahead or astern during maneuvering.

Enough has perhaps been said as to the general arrangement and estimated economy obtainable in the combined system, and now it may be of interest to refer to the general application of the system.

The development of the combination is already rapidly taking place on land, where the exhaust steam from non-condensing engines, especially the winding engines of collieries, and rolling and mill engines, is being utilized in low-pressure condensing turbines. There are, at the present time, some twenty-four of these installations of the Parsons type delivered, working and under construction, ranging from 125 to 1,250 kilo-

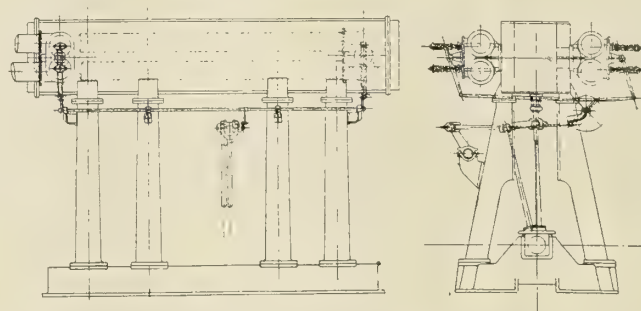


FIG. 10.—ARRANGEMENT OF DOUBLE BEAT CHANGE VALVES BETWEEN RECIPROCATING ENGINE, TURBINE AND CONDENSER, IN COMBINATION SYSTEM WORKED FROM MAIN ENGINE.

existing arrangement of machinery by introducing a reciprocating engine on the center shaft in lieu of the high-pressure turbine at present in the vessel. This engine will be of the high-speed inclosed self-lubricating type, and is now being constructed by J. S. White & Company, Cowes. It is designed to exhaust into the two low-pressure turbines at about 15 pounds absolute pressure, the cylinders of the engine being 12½, 19 and 30 inches in diameter, with a stroke of 18 inches, and revolutions about 350 per minute. It is expected that this vessel will be ready for trials in about four months' time.

THE LAYING OUT OF PROPELLER WHEELS.

BY CHARLES S. LINCH.

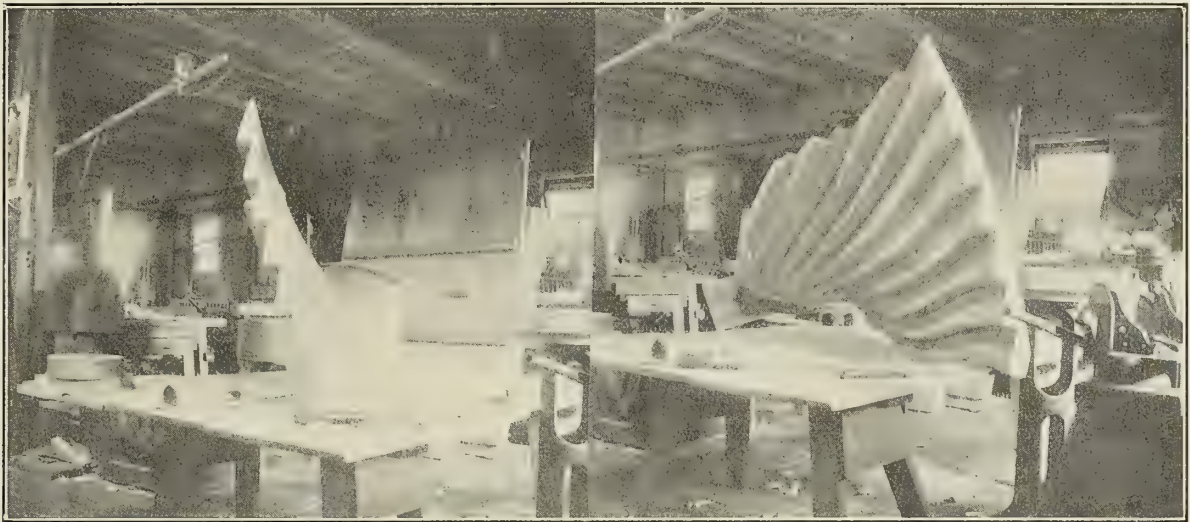
PATTERN MAKING.

We will now describe briefly the method of building up the pattern. The description, with the accompanying photographs, will make the subject clear, but to enter into every detail would far exceed the limits of these columns. We will take for illustration the propeller shown in Fig. 7. This wheel is 10 feet 6 inches diameter; 14 feet pitch; four blades.

The patternmaker first lays off on a large board to full size the view looking down on the wheel, as shown, and proceeds to lay down the sections of blades at their proper angle. This view is then divided up into a number of sections to suit the thickness of lumber, which is purely arbitrary. In this case the sections are $2\frac{1}{2}$ inches thick. Having fixed on this detail, we proceed to get out a template, as shown in Fig. 8. On this template we set off points corresponding to the divisions, as



THE PIECES ARE GLUED TOGETHER AND EDGES ROUGHLY REMOVED BY A HATCHET. THE HUB IS INTEGRAL WITH THE BLADES.



THIS GIVES A CLEAR INSIGHT INTO THE MANNER OF BUILDING UP THE SECTIONS. THE DRIVING FACE IS FINISHED, BUT THE BACK AS YET UNTOUCHED. THE BATTEN IS SPRUNG IN ON THE FACE.

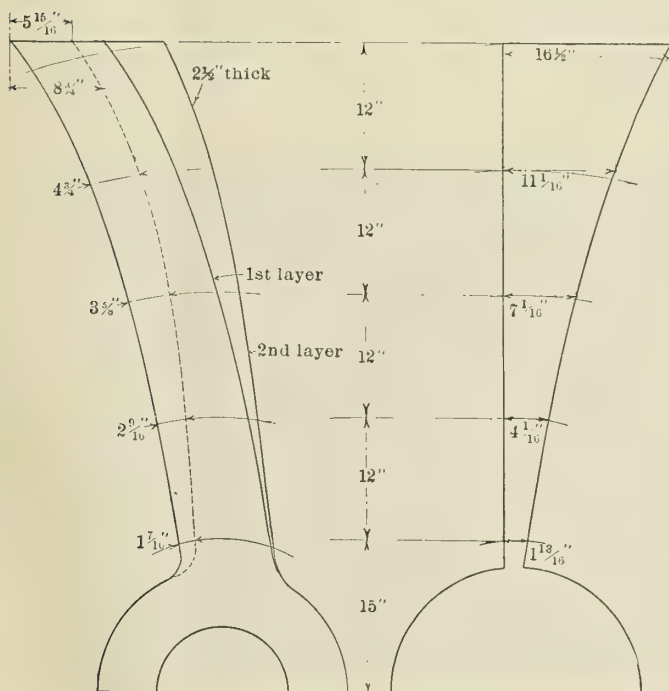
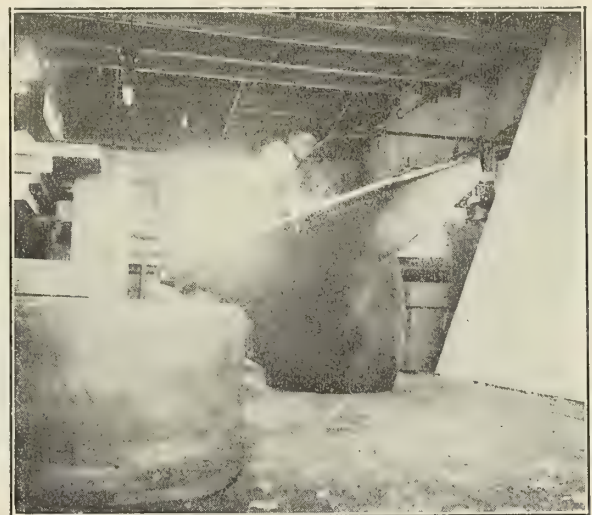


FIG. 8.



DRESSING DOWN FILLET CONNECTING BLADE AND HUB.

shown on Fig. 7. Through these points arcs are struck, as shown. It will be observed that the section of hub enters every piece that goes into the pattern, and hence forms an integral part of same, which is superior to doweling, gluing or nailing it in place. The number of pieces in this blade, as will be seen by the photographs, is eleven. Each piece is cut from the

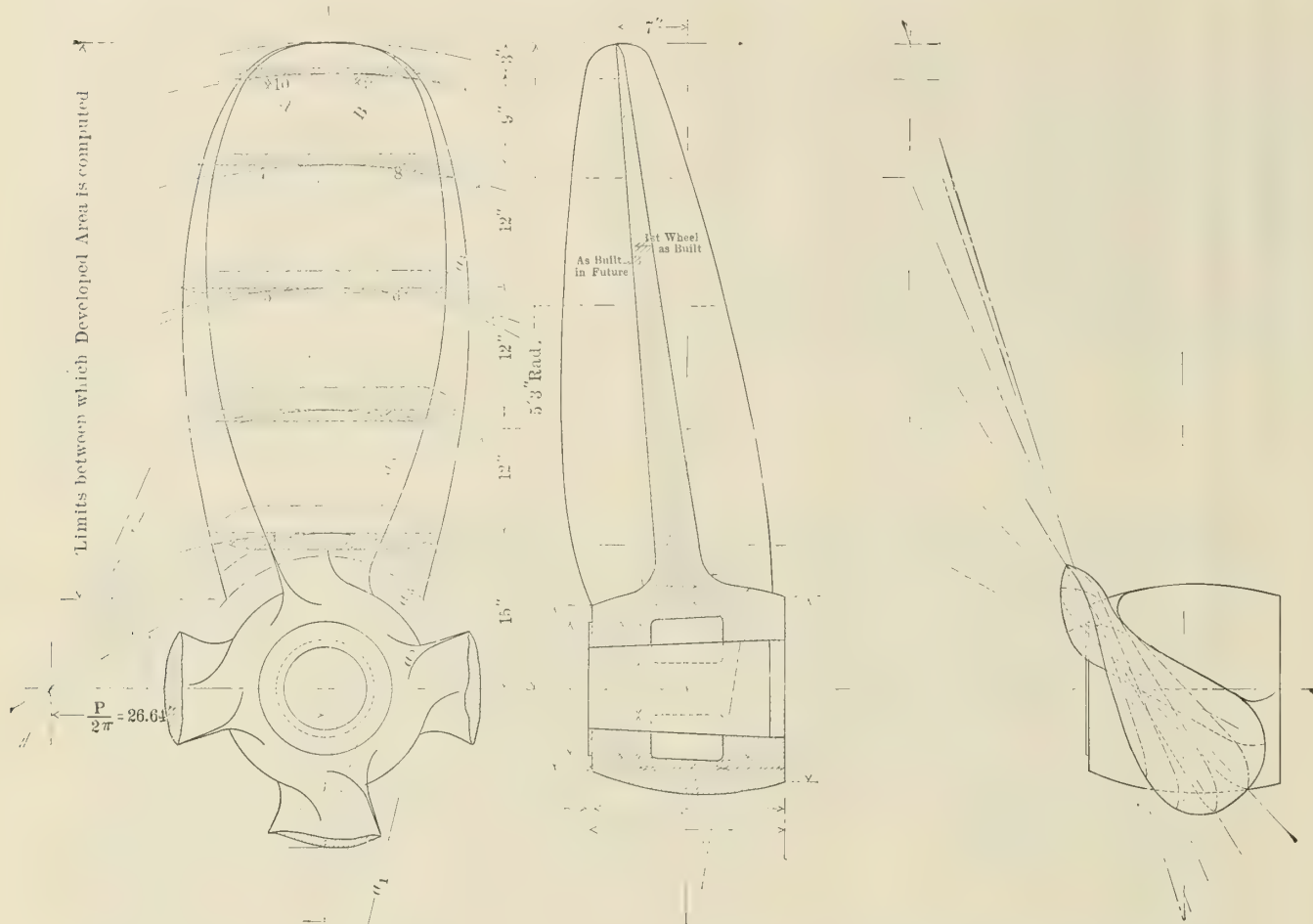


FIG. 7.—THE DEVELOPMENT OF A PROPELLER WHEEL OF 10 FEET 6 INCHES DIAMETER AND 14 FEET PITCH.

template shown. After the pieces are cut out they are glued up, as shown in the photograph. Now, on each of these arcs, as the pieces are in position, we have the cylinder intersecting the screw surface.

On each arc are driven hardwood pins, $\frac{1}{8}$ inch larger than the thickness of blade at respective sections. This enables the patternmaker to chop off the corners and rough out the face and back, after which it is dressed down to finished dimensions. In photographs are shown the edges being reduced, and the driving face worked to finished dimensions. This is finished before the back of the blade is touched. We now see that the intersections of these cylinders are helices. Further,



THE BLADE STANDING ON THE HUB, WITH BATTENS SPRUNG IN DEFINING BLADE EDGES. THE DIMENSIONS FOR CHECKING SHAPE OF BLADE ARE TAKEN BETWEEN THESE. THE TWO VIEWS AT RIGHT ARE OF THE COMPLETED PATTERN OF ONE BLADE.

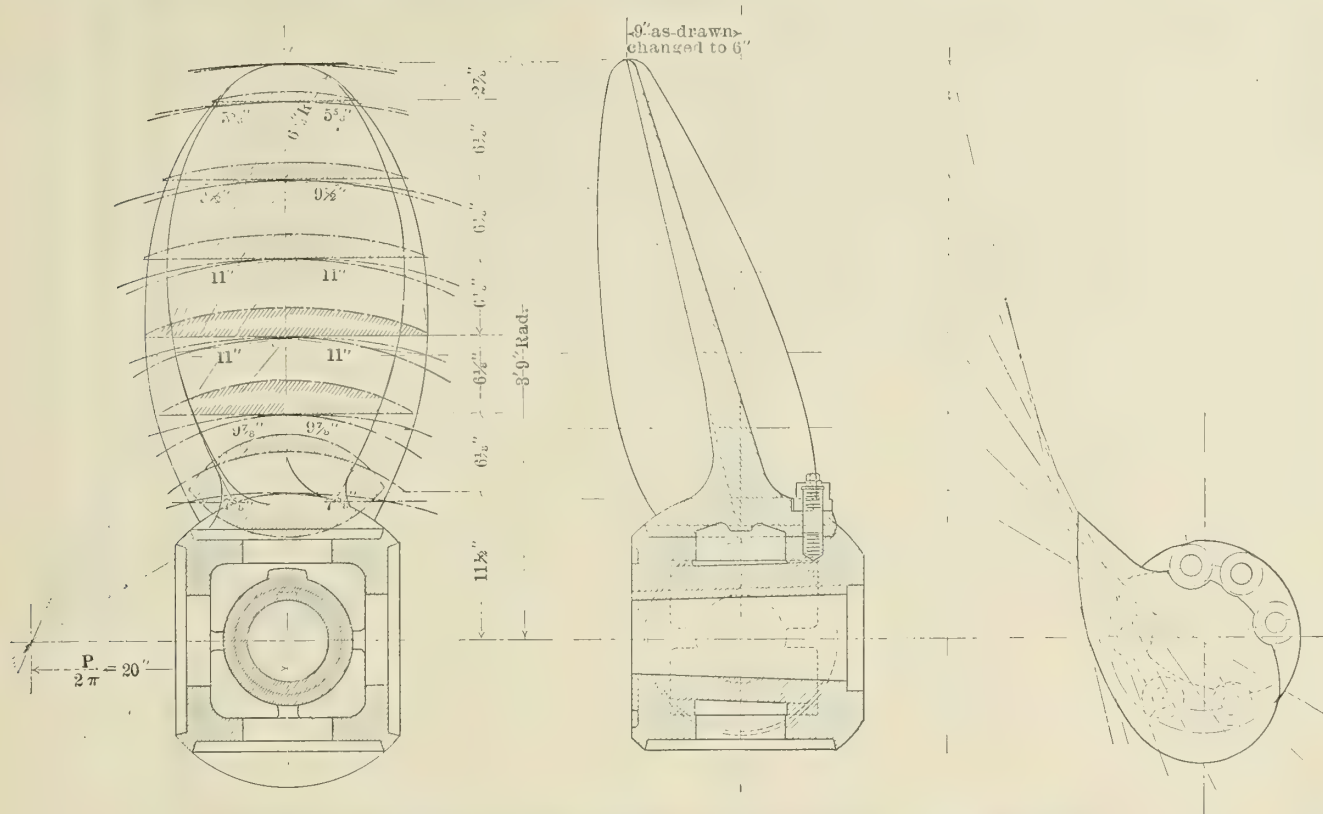


FIG. 9.—THE DEVELOPMENT OF A PROPELLER WHEEL OF 7½ FEET DIAMETER AND 10½ FEET PITCH.

we see that the length of blade is small compared with the pitch. We also note that these curves are elliptic arcs. This can be proved by drawing these arcs full size and laying them down on the pattern, showing coincidence throughout. After the face has been finished, the back is next worked down, after

which the hub is brought down to dimensions. In this method of building the pattern we observe that the fillets are formed in the piece, which again is superior to fitting the fillets after the hub and blade are finished.

As noted, after the face and back have been brought to the

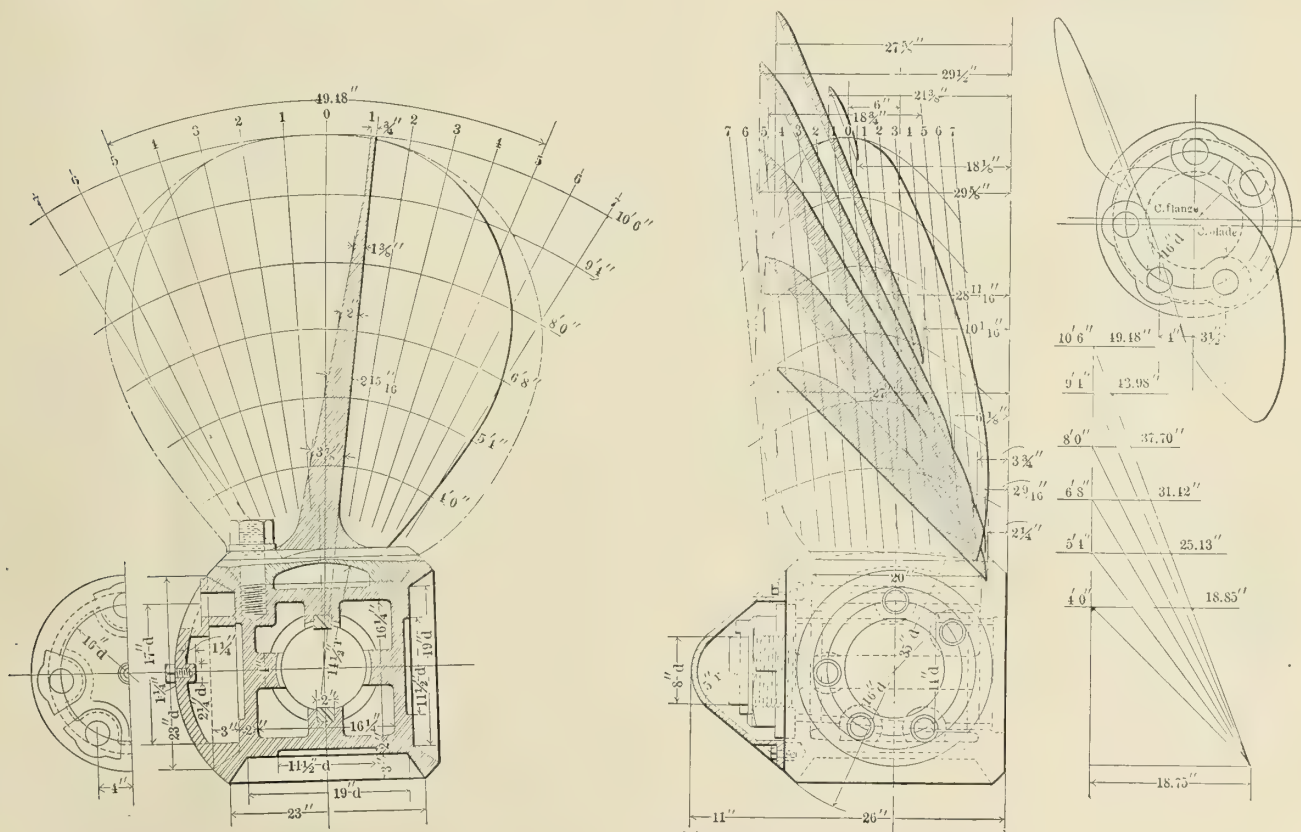
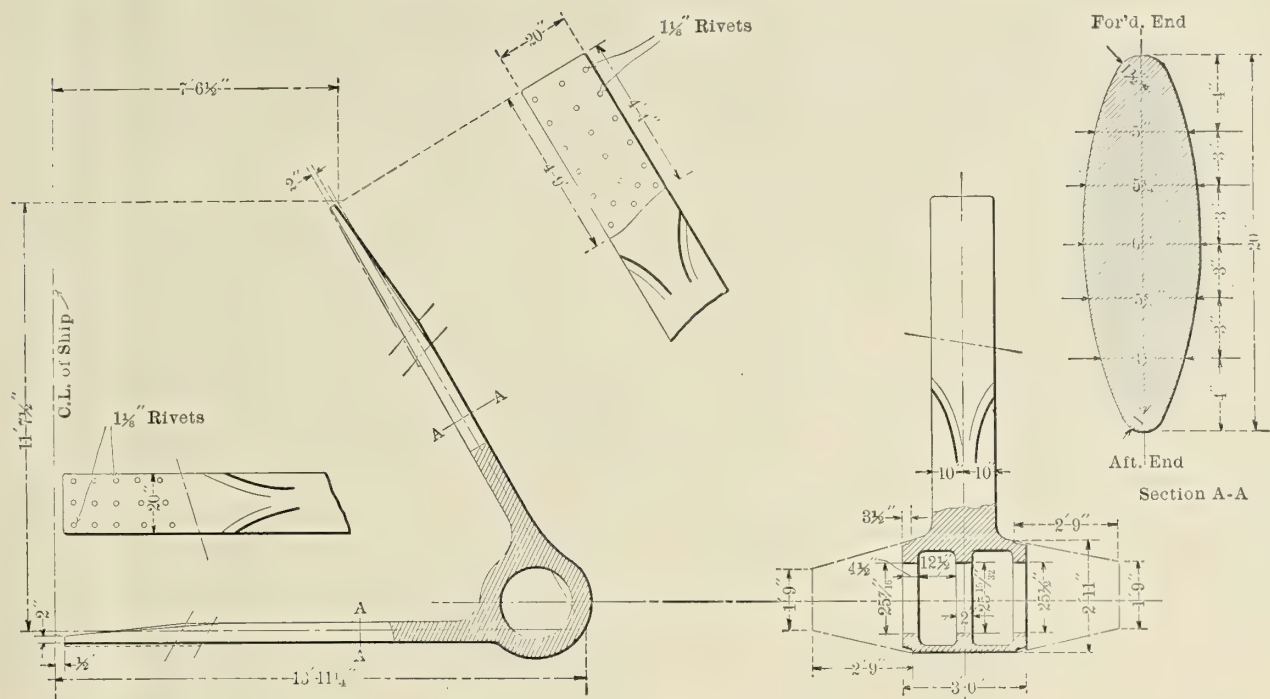


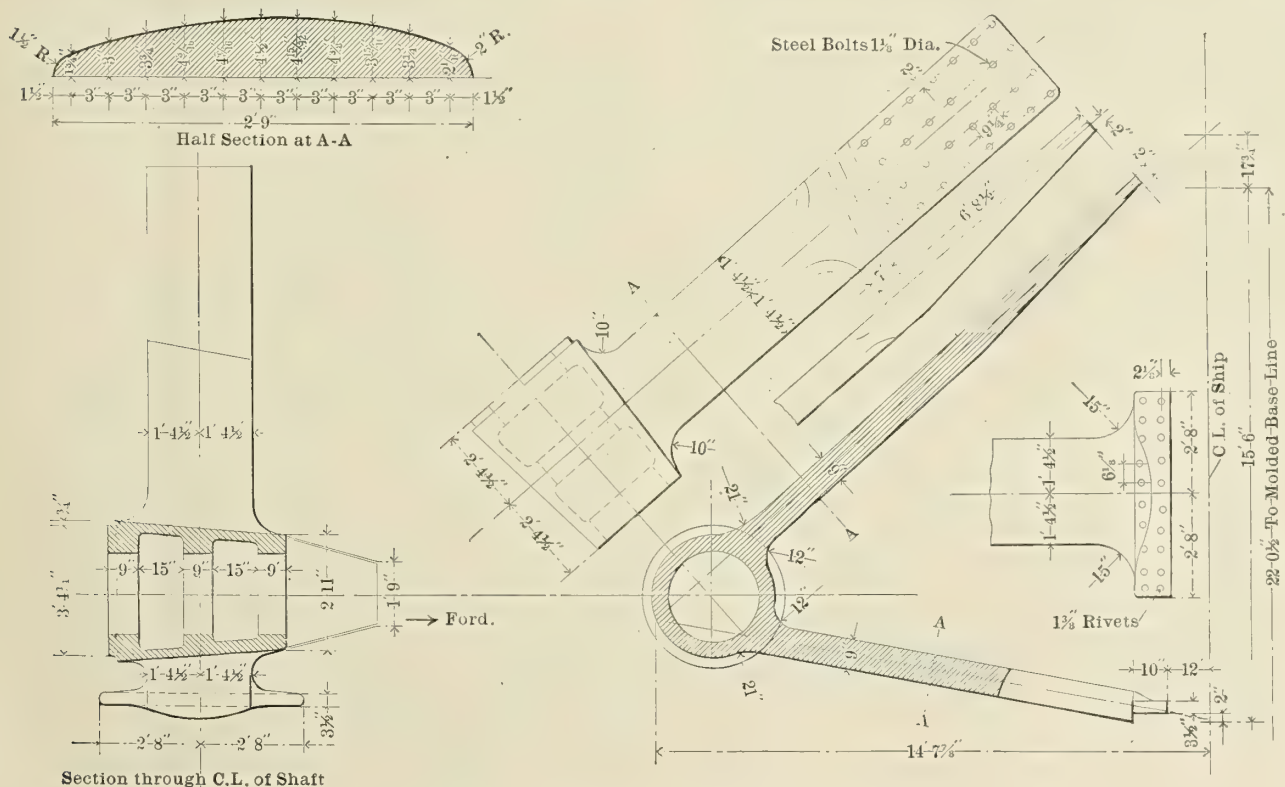
FIG. 10.—LAYOUT OF THE PROPELLER OF PEARY'S ARCTIC EXPLORATION SHIP ROOSEVELT.



each of the after struts being of 31,140 pounds, oval in section, 33 by 10 inches, and of cast steel. The center of the shaft in the strut is 8 feet 4 $\frac{1}{32}$ inches above the molded base line of the ship, and 11 feet 4 $\frac{15}{64}$ inches out from the center line. The boss has a length of 4 feet.

The inclination of the shaft is a little greater than in the *California*, the forward and downward inclination being at an angle of 45 minutes 4 seconds, or 0.1573 inch per foot, while

the forward and inward inclination is at an angle of 1 degree 28 minutes 44 seconds, or 0.3097 inch per foot. The center lines of the shaft prolonged would meet 38 feet 8 inches forward of the forward perpendicular of the ship, and 30.83 inches above the molded base line. The two propellers are of manganese bronze, weighing 29,583 pounds each. The pitch ratio is 1.132, the diameter being 17 feet and the pitch 19 feet 3 inches. Each propeller is located 4 feet 11½ inches



aft of the center line of the after strut. The speed is 22.2 knots.

In the case of the *Mexican* and *Columbian*, the center line of the arm does not intersect the center line of the shaft, there being an offset, as shown on the drawing, of 15 inches. The center lines of the arms of the struts on the *California* prolonged meet in a point considerably offset from the center line of the shaft, as shown in the drawings. The struts of the *Milwaukee* and *Ohio*, however, have arms whose center lines prolonged meet the center line of the shaft. In the *Tacoma* again, the arms are offset, their center lines meeting outside the center line of the shaft.

SPEED TRIALS AND SERVICE PERFORMANCE OF THE CUNARD TURBINE STEAMER LUSITANIA.*

BY THOMAS BELL.

The following hourly abstract on one of the watches on the *Lusitania* brings home to one's mind the loss in steam and speed caused by cleaning fires, especially when the coal is small. It can be easily calculated from this what an appreciable increase in the ship's mean speed could be obtained from this cause alone, if the price and supply would admit of the use of some system of oil-fuel burning:

	Mean Revolutions.	Corresponding Speed of Ship.
First hour.....	178	about 24. knots
Second hour.....	181	about 24.3 knots
Third hour.....	186	about 25.0 knots
Fourth hour.....	187	about 25.1 knots
Mean for watch.....	183	about 24.6 knots

Regarding the observations from readings taken in the engine room on the official trials generally, it may be stated that on the measured miles the revolutions were obtained from electric records in connection with the pallograph apparatus, but on the lengthened trials they were taken from half-hourly readings of the engine-room counters. The vacuum recorded is that of the vapor in the main exhaust orifice forming the top of the condensers, and as measured by a siphon mercury gage, the readings of which throughout are corrected to correspond to a 30-inch barometer. The total quantity of feed water is obtained from hourly counter readings of the double strokes of the Weir's feed pumps, the average length of stroke and the slip or leakage of each pump being determined, both before and after the trials, by careful tests.

The consumption of steam of the auxiliary machinery is obtained by noting the amount by which the temperature of the total feed water was raised in the feed heaters, and to the amount thus found must be added the steam used in the turbo-generators, the exhaust from which was led direct to the auxiliary condensers on the official trials. As before stated, in actual service these turbo-generators exhaust into the contact heaters, and thus raise the feed temperature to about 200 degrees. These connections had to be slightly altered at the time of the trials, and, unfortunately, therefore, advantage could not then be taken of this additional source of economy.

The torque horsepower was obtained by the Denny-Johnson apparatus, and the records show that, while a propulsive efficiency of the whole installation was obtained which accorded with the original estimate, the steam consumption of the turbines themselves was very satisfactory. It need hardly be pointed out that those two, viz.: propulsive efficiency and steam consumption per unit of power, form an excellent check on each other, for whatever would unduly favor one would be at the expense of the other.

The *Lusitania* was floated out of drydock at Liverpool on July 22, 1907, and was thereafter coaled by the Cunard Com-

pany, the bunkers for the forward and after boiler rooms being filled with South Wales coal, and those of the two middle boiler rooms with Yorkshire coal.

On her return to the Clyde, on the morning of July 27, a series of progressive runs was made on the Skelmorlie measured mile, as recorded on Table I., with the ship at a mean draft of 32 feet 9 inches, corresponding with a displacement of

TABLE I.

Time.	PRESSURES.		Vacuum at 30 In. Barom- eter.	Revs. per Minute.	Speed in Knots.	Shaft Horse- power.	Slip of Pro- pellers.
	H.P. Recr. Pounds.	L.P. Recr. Pounds.					
First double run...	157	51 pounds	28. "	194.3	25.62	76,000	17.2%
Second double run...	135	2 1/2 pounds	27.9"	186.0	25.0	65,500	15.5%
Third double run...	110	1/2 pound	28.1"	174.2	23.7	51,300	14.5%
Fourth double run...	90	3 1/2" Vac.	28.1"	161.5	22.02	40,500	14.3%
Fifth double run...	70	6 1/2" Vac.	28. "	147.6	20.4	29,500	13.1%
Sixth double run...	50	10 1/2" Vac.	28. "	131.1	18.0	20,500	13.7%
Seventh double run	35	14 1/2" Vac.	28.1"	116.1	15.77	13,400	14.6%

37,080 tons. These results are also given in graphical form in the diagram on Fig. 5, which gives curves of shaft horsepower, revolutions and slip on a common speed base. Two other most interesting curves have been added, one showing the effective horsepower determined by means of tank experiments, and the other showing the propulsive efficiency. It was intended to repeat this trial at the termination of the official trials, at a mean draft of about 30 feet. Unfortunately, however, thick weather on the morning of Aug. 2 prevented this being carried out, but the dotted curve on Fig. 5 indicates with sufficient accuracy what might have been expected.

On the evening of the 27th the *Lusitania* proceeded on a pleasure cruise around Ireland, during which consumption trials at 18, 21 and 23 knots were carried out, and, after landing the guests in the forenoon of the 29th, the vessel returned to the Clyde, making a consumption trial at 15 3/4 knots *en route*, the results of these trials being given in the first four columns of Table V. After checking the draft of ship, etc., the 48 hours' full speed continuous trial was commenced at midnight. This trial consisted of two double runs on a course of 304 nautical miles between Corsewall Point and the Longships, and the results obtained are recorded on Table II. and the last column of Table V. The mean draft at starting was 32 feet 7 inches, and at the finish about 30 feet 8 inches, the mean for the run having been 31 feet 7 1/2 inches, corresponding with a displacement of 35,600 tons.

TABLE II.

Time.	PRESSURES.		Vacuum at 30 In. Barom- eter.	Revs. per Minute.	Speed in Knots.	Shaft Horse- power.	Slip of Pro- pellers.
	H.P. Recr. Pounds.	L.P. Recr. Pounds.					
First run.....	146	2 1/2	27.9"	188.8	26.35	70,400	
Second run.....	145	2 1/2	28. "	187.4	24.3	68,200	
Third run.....	146	2 1/2	27.9"	187.5	26.3	68,700	
Fourth run.....	148	2 1/2	27.8"	187.9	24.6	68,100	
Mean of means..	146	2 1/2	27.9"	187.9	25.4	68,850	15%

The coal consumed in the 50 hours during which the engines were running at full speed was found by measurement of bunkers to be about 2,200 tons. This represents an evaporation of 10.1 pounds of water per pound of coal from 165 degrees temperature of feed, or 11.1 pounds from and at 212 degrees, and a consumption of coal for all purposes of 1.43 pounds per shaft horsepower per hour, with a rate of combustion of coal of 24.3 pounds per square foot of grate surface per hour. The number of stokers on watch was the same as in actual Atlantic service, and the air pressure in the ashpits did not exceed 3/4 inch of water column. The port evaporators were used for 10 hours of the trial; but, as the vapor from

* Read before Institution of Naval Architects, April 9, 1908.

these was condensed in the port auxiliary condenser to which the exhaust from one set of the turbo-generators was led, they were discontinued, and the make-up feed obtained from the reserve tanks for the remainder of the time.

The third trial, recorded on Table III., which was commenced in the forenoon of Aug. 1, consisted of one full-power double run between Corsewall Point and the Chicken Rock, a distance of 59 nautical miles each way, but comparison with the dotted curve on Fig. 5 shows that the tide conditions

TABLE III.

Time.	PRESSURES.		Vacuum at 30 In. Barometer.	Revs. per Minute.	Speed in Knots.	Shaft Horsepower.	Slip of Propellers.
	H.P. Recr. Pounds.	L.P. Recr. Pounds.					
First run.....	152	2½	28"	191.3	26.75	72,000	
Second run.....	152	2½	28"	191.7	26.17	72,800	
Mean.....	152	2½	28"	191.5	26.46	72,400	13.2%

during this trial give altogether too favorable a speed result. The mean draft was 30 feet 4½ inches, corresponding with a displacement of 34,160 tons. The vessel was then headed for Ailsa Craig, and carried out the specified six full-power runs between Ailsa Craig and the Holy Isle (off Arran). These latter, recorded on Table IV., which were run at a mean draft of about 30 feet 2 inches, corresponding with a displacement of 33,770 tons, give a very reliable record of power and speed at this draft, when compared with the 25.62 knots obtained on the measured mile at 32-feet 9-inch draft. On the following day weather conditions precluded any further trials, and the reversing trial and the steering and circle-turning trials were accordingly carried out on the vessel's passage to Liverpool on Aug. 26.

In a fast passenger liner such as the *Lusitania*, it is of the utmost importance that the maneuvering capabilities should leave nothing to be desired, and to demonstrate the possibilities of the ship in this respect, various trials were made, the most important being the following:

Stopping Trial.—The ship was run on the Skelmorlie measured mile at a speed of 22.8 knots, the average revolutions of the propellers being 166 per minute. On entering the mile, the engine-room telegraphs were rung to "full speed astern"; the ship was brought to rest in 3 minutes 55 seconds, the distance run being about three-quarters of a mile, or about six times the length of the ship. During this trial the boilers in the three

TABLE IV.

Time.	PRESSURES.		Vacuum at 30 In. Barometer.	Revs. per Minute.	Speed in Knots.	Shaft Horsepower.	Slip of Propellers.
	H.P. Recr. Pounds.	L.P. Recr. Pounds.					
First run.....	151	2½	28"	191.3	25.62	72,500	
Second run.....	152	2½	28"	191.1	26.36	72,000	
Third run.....	153	3	28"	191.9	25.31	72,000	
Fourth run.....	147	2½	28"	191.0	26.16	72,100	
Fifth run.....	149	2½	28"	190.2	25.26	70,800	
Sixth run.....	149	2½	28"	191.2	25.95	71,800	
Mean of means..	150	2½	28"	191.2	25.77	71,910	15.3%

after boiler rooms only were in use, and the initial pressure at the astern turbine was about 90 pounds per square inch.

Circle Trials.—With the ship initially on a straight course, and the turbines running at an average speed of 180 revolutions per minute, the steering wheel was put hard over in 17 seconds. The tiller went over to 35 degrees in 20 seconds, and the vessel made a complete circle in 5 minutes 50 seconds, the average revolutions coming down at the completion of the circle to 70 percent of the rate at the commencement. The resulting circular path was approximately 1,000 yards, or four lengths of the ship, in diameter. This maneuver was made

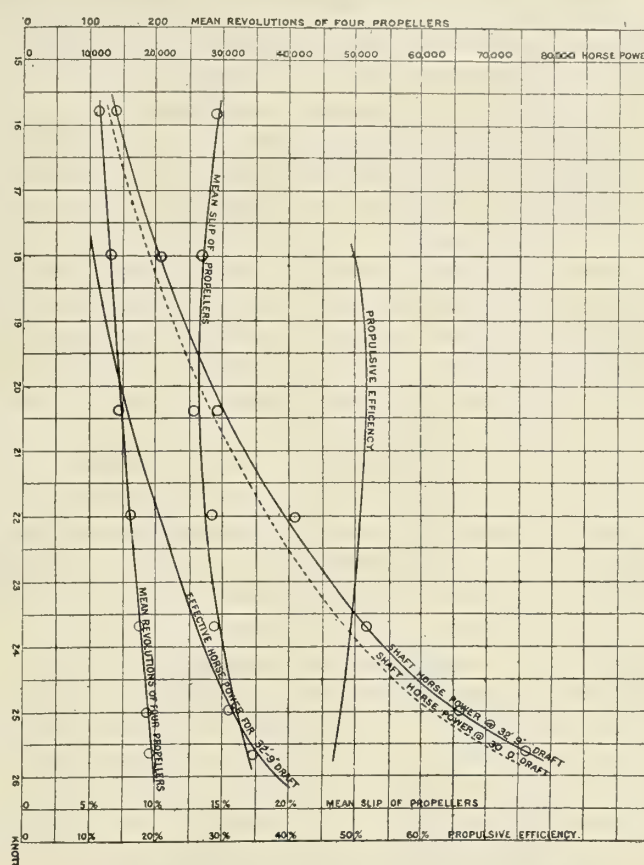


FIG. 5.—PROGRESSIVE TRIAL, JULY 27, 1907, OFF SKELMORLIE.

both under starboard and port helm with very closely confirmatory results.

Going astern with the inner propellers running at a uniform rate of 136 revolutions per minute and under full helm, resulted in half circles being made in an average time of 6 minutes 45 seconds.

As important factors contributing to these very satisfactory results, it may be remarked that, following on the suggestion of Sir Philip Watts, the deadwood aft is cut away in a fashion similar to that in recent warships. The inner propellers are fairly close together, and as the rudder is of large dimensions in the fore and aft direction, the race from these propellers impinges fully upon it when any helm is used. The vessel, consequently, is very similarly circumstanced to a single screw ship, or a triple screw ship with all three propellers in action, and gets steerage way without any perceptible headway, and this feature was very noticeable during the steering trials. At first sight it would appear that the outer propellers should have

TABLE V.

Actual steam and coal consumption of main and auxiliary engines at various speeds under conditions prevailing on official trials, viz., turbo-generators exhausting to auxiliary condensers, other auxiliaries exhausting to heaters.

	13,400	20,500	33,000	48,000	68,850
Shaft horsepower.....	15.77	18.0	21.0	23.0	25.4
Speed in knots.....	200°	200°	199°	179°	165°
Temperature of feed water.....					
Total consumption of auxiliaries in pounds per hour.....	71,000	76,400	85,700	96,700	116,500
Total consumption of turbines in pounds per hour.....	284,500	353,600	493,300	668,300	879,500
Steam consumption of auxiliaries in pounds per turbine horsepower hour.....	5.3	3.72	2.6	2.01	1.69
Steam consumption of turbines in pounds per horsepower hour.....	21.23	17.24	14.91	13.92	12.77
Total steam consumption in pounds per horsepower hour.....	26.53	20.96	17.51	15.93	14.46
Coal consumption in pounds per horsepower hour.....	2.52	2.01	1.68	1.56	1.43
Estimated coal consumption in tons on a voyage of 3,100 nautical miles, allowing 20 tons for galleys, etc.....	2,980	3,190	3,670	4,520	5,390

TABLE VI.

Estimated steam and coal consumption at various speeds, allowing for the additional auxiliary steam consumption found requisite under actual service conditions for the washing water supply, etc., with a full complement of passengers, weather conditions being as on official trial.

	13,400	20,500	33,000	48,000	68,850
Speed in knots.....	15.77	18.0	21.0	23.0	25.4
Temperature of feed water.....	200°	200°	200°	200°	200°
Total consumption of auxiliaries in pounds per hour.....	93,500	100,900	112,700	127,500	149,700
Total consumption of turbines in pounds per hour.....	284,500	353,600	493,300	668,300	879,500
Steam consumption of auxiliaries in pounds per turbine horsepower hour.....	6.97	4.92	3.41	2.65	2.17
Steam consumption of turbines in pounds per horsepower hour.....	21.23	17.24	14.91	13.92	12.77
Total steam consumption in pounds per horsepower hour.....	28.2	22.16	18.32	16.57	14.94
Coal consumption in pounds per horsepower hour.....	2.76	2.17	1.8	1.62	1.46
Estimated coal consumption in tons on a voyage of 3,100 nautical miles, allowing 20 tons for galleys, etc.....	3,270	3,440	3,930	4,700	5,490

been those utilized for maneuvering purposes, as the outer shafts have about three times the spread from the middle line that the inner shafts possess. For turning with propellers alone without the help of the rudder they would have been much the more effective, but they would not have possessed any such advantage as that alluded to above in respect to obtaining steerage way without headway.

Table VI. has been compiled for comparison with Table V., to show the additional consumption of steam for auxiliary purposes under actual working conditions at sea with the ship full of passengers. This shows very clearly the demand which modern improvements make on the steam, and hence coal consumption of a large passenger vessel. An additional line has been added to Tables V. and VI. to show total coal consumption on a voyage of 3,100 nautical miles at the various speeds and under the different conditions.

With reference to the third voyage west, from Nov. 2 to Nov. 8 of last year, thanks to the courteous permission of the chairman of the Cunard Company, the leading particulars of the official engine-room log are summarized in Table VII. Regarding the mean draft of the vessel at sea, it may be remarked that, after the second day out, certain of the forward tanks were gradually filled for the purpose of avoiding excessive trim, so that the mean draft on Nov. 5, 6 and 7 was approximately 32 feet, or very little more than the mean of the first pair of runs from Corsewall Point to the Longships and back. The conditions, however, were otherwise very different, for, with the exception of the 12 hours of fine weather and smooth sea from noon till shortly after midnight on Nov. 6, it was throughout the average mid-Atlantic winter weather—namely, strong winds and resulting boisterous sea. Up till midnight on the 6th, *i. e.*, for 2,176 out of a total of 2,781 nautical miles, the mean speed works out at 24.65 knots; but, unfortunately, early on the 7th the wind freshened, gradually increasing to a furious southwest gale, which reached its height about 4 P. M., and reduced the average speed for the last 24 hours below 23 knots, and thus brought down the mean

TABLE VIII.

Date—1907.	Length of Steam-ing Day.	Distance Run, Nautical Miles.	Speed, Knots.	Total Distance Steamed.	Total Time.	Mean Average Speed.
	Hrs.Min.				Hrs.Min.	
Noon, November 3.....	0 52	21	24.24	21	0 52	24.24
Noon, November 4.....	24 57	606	24.28	627	25 49	24.27
Noon, November 5.....	25 2	616	24.6	1,243	50 51	24.44
Noon, November 6.....	24 55	618	24.8	1,861	75 46	24.57
Noon till midnight, November 6.....	12 30	315	25.2	2,176	88 16	24.65
Noon, November 7.....	12 22	295	23.85	2,471	100 38	24.55
Morning, November 8....	14 2	310	22.09	2,781	114 40	24.25

average for the completed voyage to 24.25 knots. Table VIII., giving the mean average speeds at the different stages of the voyage, shows very clearly the effect of this gale, unfortunate so far as preventing the vessel from complying with the contract conditions, but giving those connected with the ship an opportunity of thoroughly satisfying themselves as to her behavior when driving through the huge waves at about 22½ knots, without any racing of engine or sign of laboring, and dispelling the idea, current in some minds, that turbine-propelled ships do not show to advantage in heavy weather.

The following particulars of the steam consumption are given in conjunction with the figures of coal consumption set forth in Table VII. Throughout the voyage a careful record of the feed pump counters gave an average of 998,000 pounds of water pumped into the boilers per hour. Of this, about 114,000 pounds was used by auxiliary machinery exhausting into the feed heaters, 26,000 pounds by the evaporating plant supplying feed make-up and washing water, and about 6,500 pounds for steam to the thermotanks, galleys and pantries, both of which latter figures are based on data obtained from tests carried out before the vessel left the Clyde. Hence, taking the average shaft horsepower as 65,000, the steam consumption per shaft horsepower per hour works out as follows:

	Per Shaft Horsepower Hour.
Total Water.....	
Main turbines.....	851,500 pounds. = 13.1 pounds.
Auxiliary machinery.....	114,000 pounds. = 1.75 pounds.
Evaporating plant and heating.....	32,500 pounds. = .5 pounds.
	998,000 pounds. 15.35 pounds.
Average amount of coal burnt per hour for all purposes =	43½ tons.
Water evaporated per pound of coal = 10.2 from a feed temperature of 196°.	
Water evaporated per pound of coal = 10.9 from and at 212°.	
Coal for all purposes per shaft horsepower per hour =	1.5 pounds.
Coal per square foot of grate per hour.....	24.1 pounds.

Taking a mean displacement of 36,000 tons, this represents at 24½ knots a consumption of almost exactly 11 pounds of coal per 100 nautical miles per ton of displacement. The coal used was half South Wales and half Yorkshire, practically the same as on the official trials.

TABLE VII.

ABSTRACT OF ENGINE-ROOM LOG FOR THIRD VOYAGE WEST: QUEENSTOWN TO NEW YORK.
Date when last dry docked July 22, 1907. Mean draft, leaving Queenstown, 33 feet 7 inches. Mean draft, arriving New York, 30 feet 10 inches.

Date 1907.	STEAM PRESSURES.			TEMPERATURES.		Vacuum.	Baro-meter.	LENGTH OF DAY.		Distance by Observation, Nautical Miles.	Mean Speed.	Mean Revs.	Mean Slip Percent.	Coal Consumed for Main and Auxiliary Engines.
	Boilers. Pounds.	H.P. Recrs. Pounds.	L.P. Recrs. Pounds.	Hot Well.	Feed Water.			H's	M's					
Noon, November 3.....	170.	140.0	2.3	68°	200°	28"	30.4"	24	52	21	24.24	182.5	16.5	40 tons
Noon, November 4.....	169.1	142.2	2.2	78°	197°	28"	29.7"	24	57	606	24.28	182.6	16.4	1,090 tons
Noon, November 5.....	167.3	140.6	2.3	78°	198°	28.2"	30"	25	2	616	24.6	182.8	15.4	1,090 tons
Noon, November 6.....	168.3	140.4	2.5	70°	196°	28.2"	30.1"	24	55	618	24.8	183.5	15.1	1,090 tons
Noon, November 7.....	168.3	138.3	2.2	72°	195°	28"	29.6"	24	52	610	24.52	181.4	15.0	1,090 tons
1.14 a.m. November 8.....	165.	132.5	1.5	75°	200°	27.8"	29.3"	14	2	310	22.09	174.	20.2	*576 tons
Means.....	168.	139.3	2.2	74.5°	197°	28.1"	29.8"	Total.	Total.	2,781	24.25	181.1	15.9	4,976 tons
								114	40					

* This includes all coal used till 10 a. m. on the 8th.

Summary of total coal consumed on voyage:—Liverpool to Queenstown, 408 tons; Queenstown to New York, 4,976 tons; galleys, etc., 18 tons; total coal taken from bunkers, from leaving landing stage, Liverpool, till moored at wharf, New York, 5,402 tons. Passage—Queenstown to Sandy Hook—4 days, 18 hours, 40 minutes.

THE BOSTON FLOATING HOSPITAL.

BY ROBERT CHARLES MONTEAGLE.

The Boston floating hospital for infants and small children, as it exists to-day, is the development of fourteen years of study and actual practice in the wants of thousands of little sick children. The work in Boston began in 1894, when five experimental day trips were made on a hired barge, with volunteer service only. The following year, under the same system, thirteen day trips were undertaken, and in 1896 daily trips were inaugurated. The work was then given into the hands of a board of managers—a medical staff and two paid nurses. In 1897 the barge *Clifford* was bought, and equipped to care for 150 sick infants and children, she being towed during these trips. In 1898 two wards were established for permanent patients, and a night service was organized. From 1898 to 1900 the work developed steadily, and has continued to develop until the need for a larger vessel became a necessity. In the fall of 1905 it was decided to build a new vessel. This has been done, and in August, 1906, the new vessel made her first trip.

and 10 feet 8 inches molded depth. The gross tonnage is 594. The hull is of steel, and was designed to have the most simple construction with the maximum accommodations, without regard to speed—this latter being of no moment. Her plating is 12 pounds, the sheerstrake being 18 and 20 pounds; frames, 3 by 5 inches by 11 pounds, spaced 30 inches, and bracketed to the deck beams. A number of deep floors are placed in the machinery space and at other locations. The deck beams are 3 by 5 inches by 11 pounds, placed on every frame. Three plate and angle stringer bars run the full length of the vessel under the deck beams, as shown in the midship section. Deck stringers are 30 inches by 18 pounds. The stem is a steel bar, 1¾ by 6 inches, 12 feet high.

The vessel has six watertight bulkheads, three of these being fitted with watertight doors. Fenders are placed to best advantage in case of the vessel being handled by towboats, as she was during her first season. Double rudders are fitted, operated by steam steering gear. The bottom and sides below the floor are covered with Portland cement, thick enough to cover the rivet heads. The kitchen floor is of concrete, finished on top with pure cement, as is also the floor under



THE BOSTON FLOATING HOSPITAL PROCEEDING TO SEA WITH A CARGO OF SICK INFANTS.
Photograph, N. L. Stebbins.

The new *Boston Floating Hospital* was especially designed by Burgess & Packard, Marblehead, Mass., and built by the Atlantic Works, East Boston, as a completely equipped modern hospital—embodying the results of these fourteen years of progressive study, and embracing the best thought of the brightest minds in the medical profession. The designers of this vessel had a difficult task—out of the numerous ideas suggested—to embody the worthy and reject the unworthy. In conjunction with the builders of the vessel this they succeeded in doing marvellously well. The photograph and drawings here presented are sufficient evidence of this, and should further proof be desired, a visit to the vessel—an interview with any of those connected with the work—will be found convincing. But the most convincing of all arguments is to witness the hundreds of poor, helpless, sick children and their mothers as they troop aboard, and then note the contrast in their appearance after breathing the medicine of God's pure sea air for even one day, or, in more desperate cases, for a number of days.

The new vessel, which was launched July 7, 1906, is 171 feet long over all, 165 feet on the waterline, 45 feet molded beam,

the refrigerators; 12-inch air ports are fitted. The deck is of yellow pine, 3 inches thick. The guard is formed of a thickness of 7 by 12-inch yellow pine, and a thickness of white oak, 5 by 12 inches, going completely around the vessel. Three wood keels, 24 by 12 inches, are fitted. These are formed from 12 by 12-inch and 9 by 12-inch yellow pine, with 3 by 12-inch white oak shoes.

The material for the joiner work was selected sound and free from defects and well seasoned. All moldings and corners are rounded, and all surfaces smoothed; the aim being to keep all work free from beading and crevices or depressions which would catch dust or disease germs. Especial care was taken to use nothing but well dried stock, which would not shrink and leave cracks for the shelter of dirt and the propagation of bacteria. Particular care in this respect was given to the hospital wards, and most particularly of all to the babies' food rooms. The decks are ¾-inch matched white pine in long lengths, covered with No. 10 duck, attached with galvanized tacks. Deck beams are 1¾ by 3¾-inch yellow pine, spaced 20 inches between centers, extending full width and in one piece. There are three fore-and-aft stringers of yellow

pine, one on center line of ship, the others being spaced 10 feet off center. Stanchions are spaced about 10 feet, fore and aft. The joiner hardware is all of brass.

With a view to the efficiency of the artificial ventilating plant, great care in making air-tight work was observed in the construction of all wards and the other various rooms throughout the vessel, this latter being absolutely necessary in the case of the air-filtering rooms and the morgue. The great importance of keeping the wards free from flies or other insects was realized, and fly screens are fitted around all windows and in doorways. The outdoor ward aft is fitted with these screens completely around; all screens being made removable for storage during the winter months.

The hold contains the chain locker, crew's quarters, laundry, air-filtering room, morgue, formalin room, coal bunkers, boiler room, engine room, mothers' dining room and lecture room, kitchen and refrigerating rooms.

On the main deck are the executive office, resident physician's office, two ante-rooms, one large ward, food laboratory, sterilizing room, pharmacy, clinical laboratory, sewing and linen room, linen closet, nurses' locker and toilet room, doctors' and nurses' dining rooms, men's lockers and toilets.

On the hospital deck are six large wards, six ante-rooms, and baths and sterilizing room.

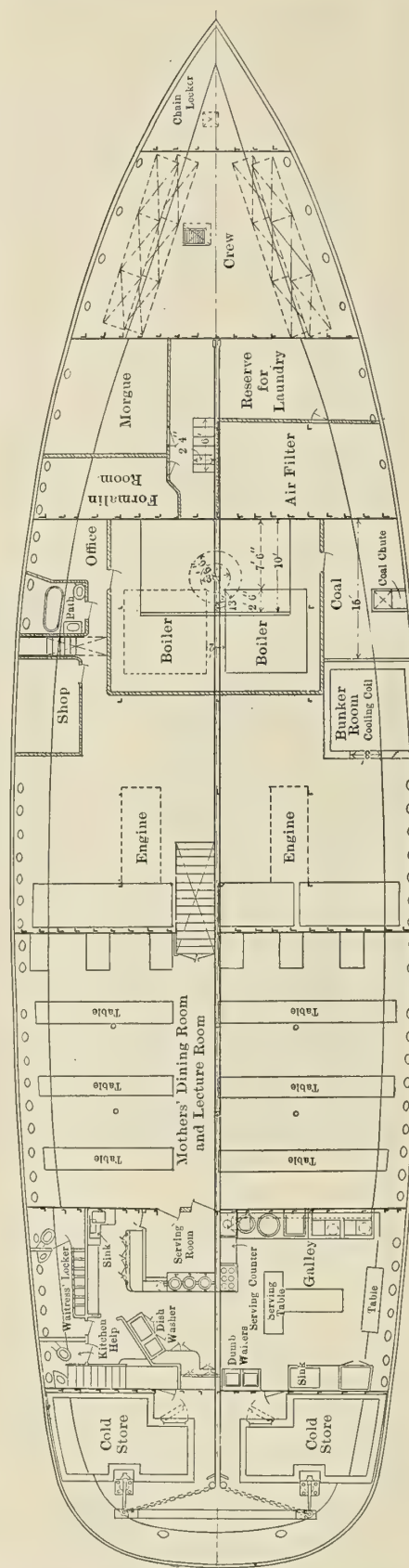
On the day patients' deck are the out-door kindergarten, nurses' resting room, assistant manager's room, the mothers' resting room, shower bath, dressing room and toilets, day patients' examination and treatment room, six staterooms for doctors, resident physician's stateroom, room for visiting physicians' staff and library, doctors' bath and toilet. On the day patients' deck is ample open air space for 150 beds and seating accommodations for the mothers.

The seven permanent wards have a capacity of 150 beds. The ward situated at the stern, on the hospital deck, is an out-door ward, where patients remain in the open air both day and night. This ward is fitted with steam heat, so that the temperature may be partially modified, should occasion require.

On the hurricane deck are the pilot house, captain's and engineer's rooms, boats, etc. This deck is not supposed to be used by the patients, but may be should occasion demand.

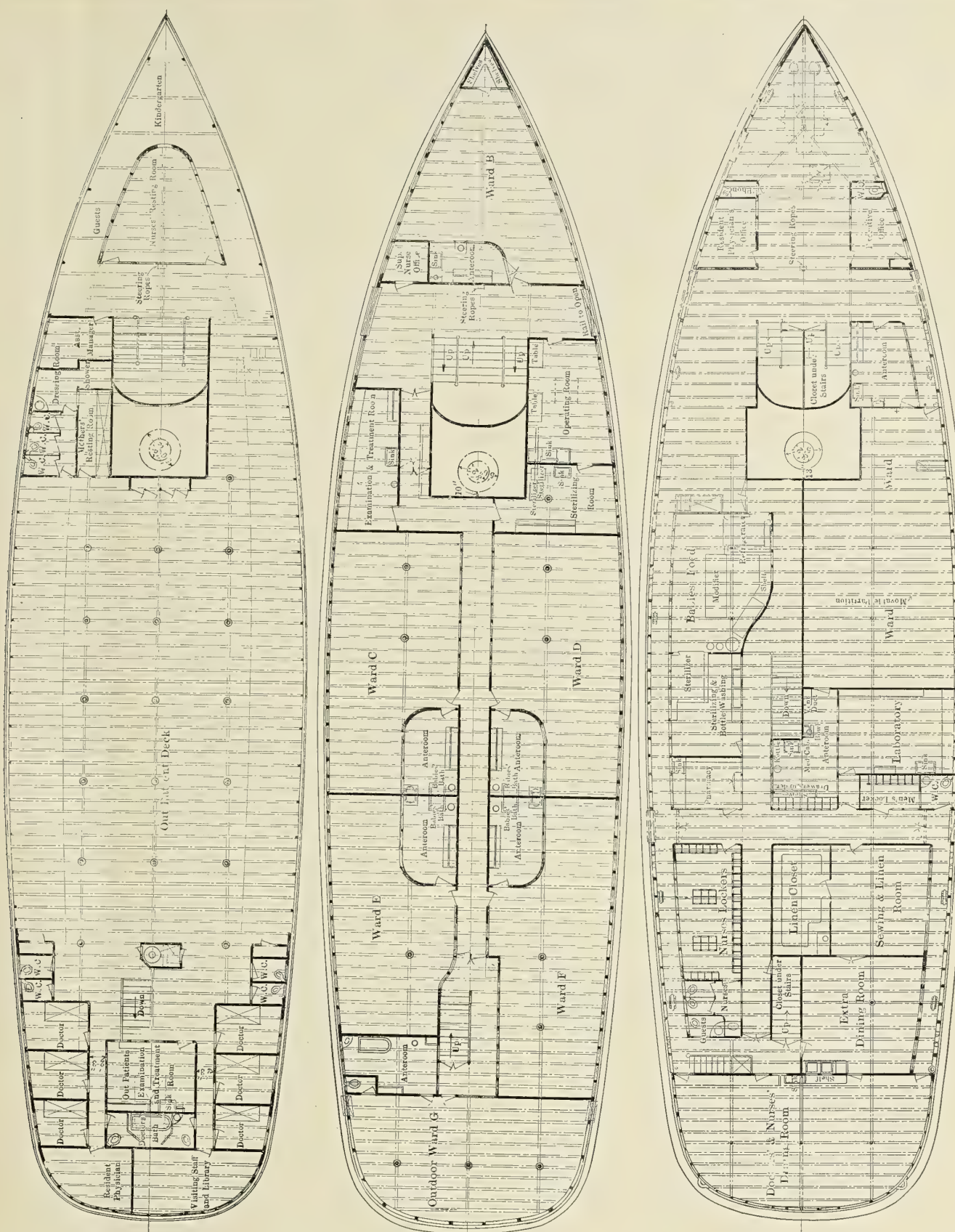
Forty to fifty nurses attend to the needs of the little ones during the daily trips of the vessel down the harbor. The floating hospital leaves her wharf at 9 A. M. and docks again at 4.30 P. M. Sick children not requiring permanent treatment, or those for whom room cannot be found in the permanent wards, are admitted to the day patients' deck, accompanied by their mothers, and are given the benefit of the day trip. During this time the patients receive proper diet and the attendance of doctors and nurses, who also instruct the mothers in nursing and hygiene. Many of the mothers, who may be worn out with anxiety or home burdens, are greatly benefited by the fresh air, excellent food, and the knowledge that their little ones may be restored to health. Any child under six years of age, and who is not suffering from a contagious disease, is admitted. Careful examination of each applicant is made on the wharf before a patient is passed.

An annual post-graduate course of instruction for trained nurses, extending from July 1 to Sept. 15, is an excellent feature connected with this institution. It includes practical experience in bedside nursing of infants and small children, with all diseases incident to early life. The visiting staff of doctors lecture annually on infant feeding, premature and sickly infants, diseases of gastro enteric tract, observations of symptoms in sick and well children, defects and deformities, diseases of and care of the skin, feeble minded and backward infants, and materia medica. Diplomas are given to those who pass creditable examinations in these lectures, and who prove themselves capable in ward work. Board and rooms are



furnished by the hospital and \$3.00 (12s.) per week remuneration for personal expenses and laundry is allowed.

It has been the aim of the trustees to establish the hospital upon a business basis, and this they have succeeded in doing. As this is realized by the public, funds are more freely donated for the support of this excellent work. The hospital cares for the sick children of the poor during the hot summer months,



THE BOSTON FLOATING HOSPITAL—OUT-PATIENTS' DECK; HOSPITAL DECK; MAIN DECK.

at a time when children's diseases are most prevalent, and when some of the other hospitals for infants are closed. There is no other similar institution in the world with the scope of this one. Other cities have floating hospitals, but when their equipment is considered, and their limited powers for help to poor children are compared with those of Boston's institution, one is given to wonder why other vessels of this

character have not found concrete expression at an earlier date.

Statistics of the cases treated by the Boston floating hospital, all involving children under six years of age, demonstrate the service being rendered to the community. A very large percentage of these cases are desperate when received. Children given up by physicians are often recommended to the

floating hospital as the "dernier resort." During last season fifty deaths occurred within forty-eight hours from the time of admission. They were in reality moribund when they were admitted. In the opinion of competent physicians, most of the children who were admitted to the permanent wards, and who had a fighting chance for life, were suffering from disorders under which, had they been left in their sweltering homes and unhygienic surroundings, they would have speedily succumbed.

When there is room for them, well children, whom their mothers, accompanying sick children, cannot leave at home, are permitted to make the day trip. These happy children sit out on the forward deck and play games under the guidance of a trained kindergarten teacher. The benefits of the education imparted to the mothers are incalculable. It is a fact that some ignorant parents give to their tiny offspring pickles, green apples, raw onions, cheese, black coffee, strong tea, beer, wine and whiskey. Such indulgences, it is needless to state, are "tabooed" on the floating hospital, and the mothers are instructed that such food is absolutely unfit for children, and if persisted in will probably result in death.

The food laboratory of this vessel has already made very important contributions to our present-day knowledge of infant feeding. It has now attained a quality of milk which is as near perfection as possible. The ideal milk for infants is naturally that of the mother. The aim has been to imitate the lactic mixtures of human milk as closely as possible. Cow's milk, it has been found, contains about four times as much curd as does human milk. In its natural form it is therefore not a suitable food for babies. To make the milk more easily digestible has been the aim of the chemists of the Boston floating hospital, and it is one which they have solved. At the same time, perfection practically has been reached in the sterilization of milk. The Boston Board of Health sets the limit of bacteria allowable per cubic centimeter as 500,000. On the Boston floating hospital this is reduced to a bare 150. Absolute cleanliness is insisted upon, and disease germs are banished to all intents and purposes.

An interesting and highly important department of the vessel is the kitchen. The range is 7 feet long by 39 inches wide, with two fires equipped with revolving grates, and two ovens 21 by 28 inches. A double shelf 18 inches wide runs the whole length of the range. There is one 35-gallon jacketed kettle. One "bain marie," or apparatus for keeping food at a constant temperature, is of heavy copper, and contains four 10-inch porcelain soup and vegetable jars with polished metal covers, and two meat pans with copper covers. The water is heated by means of a steam coil.

There are also two 30-gallon heavy vegetable steamers, with patent water seal extending around the entire top of the steamer; one heavy galvanized iron dishwasher, 62 by 24 inches, with washing and rinsing tank, and containing patent rocker for dish basket, operated by a hand lever (the dishwasher is heated by a steam coil); one 5-gallon chocolate urn; one 5-gallon coffee urn, and one 10-gallon water urn, all furnished with gages, safety valves, muslin percolators and clean-out faucets; one milk cooler, made with double walls, and dead air space between walls. In this cooler there is a heavy cast-iron jar with porcelain lining and cover, being held in place by means of a perforated shield. There is one 7-foot triple bar pot rack, with necessary hooks and hangers; one pot sink, measuring 36 inches by 24 inches by 14 inches deep; one 14 by 10-inch glass and silver sink; seven 8-inch copper sterilizing kettles, with steam jackets and tinned linings; one hood of sheet steel extending over the range, vegetable steamers and kettle.

The vessel is equipped with a modern system of plumbing, adapted especially for its peculiar service. Hot and cold, fresh and salt water are piped throughout the vessel. The

fresh water supply is taken from ten large tanks placed under the floors, eight of these being under the mothers' dining-room, and two forward of the boiler room. The drinking water system, which is also piped all over the vessel, is taken from a large tank in the engine room provided with a brine coil for cooling. The hot-water system is taken from the engine room, steam coils there furnishing the necessary heat.

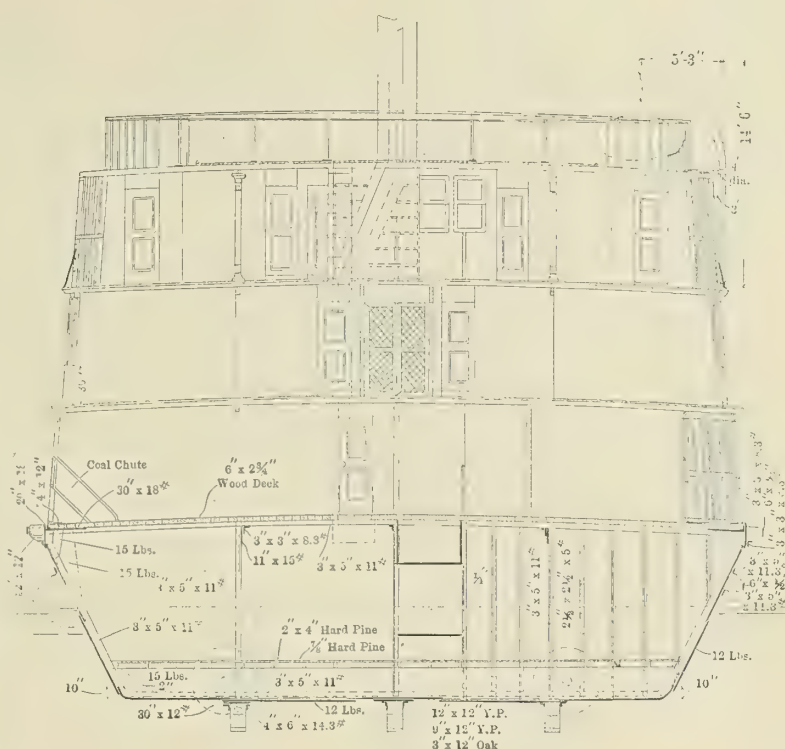
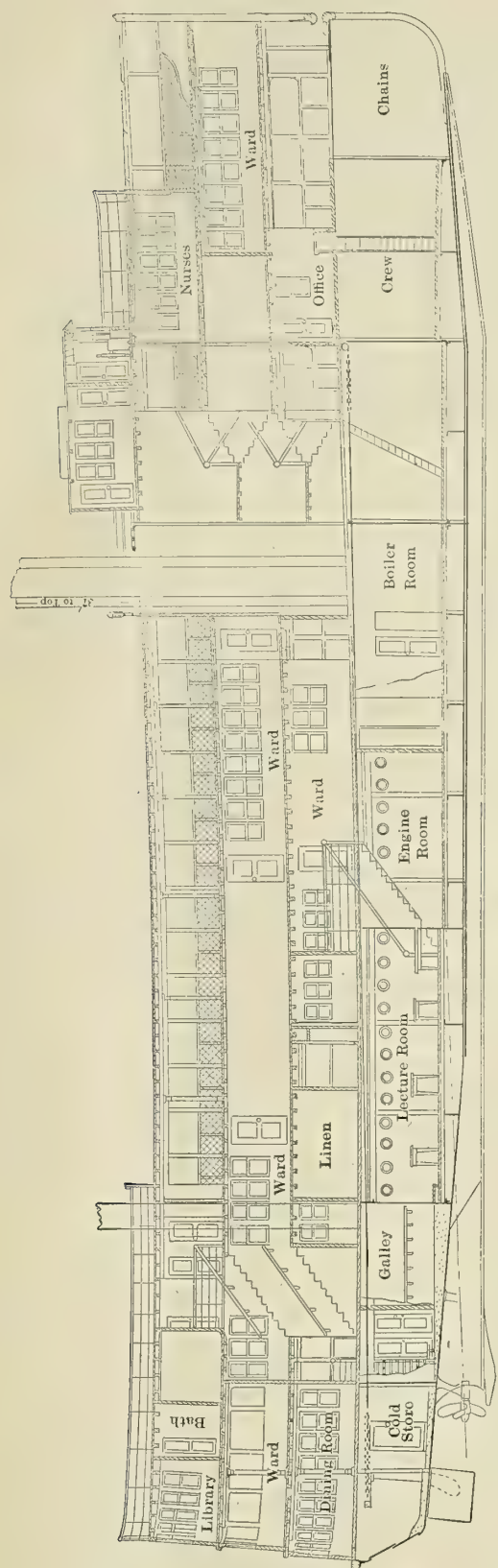
Particularly interesting and necessary features of the plumbing are the infants' baths. They are of porcelain, 24 by 20 inches by 12 inches deep, fitted with traps, hot and cold water supply and waste valves. The hot and cold water supply, as well as the waste, are operated by knee action, thus making it perfectly convenient for the nurse to have full use of her hands in attending to the child.

A complete fire system is distributed over the various decks, the steam fire pump being always connected up and ready for duty. In addition to the steam pump are two hand fire pumps connected into the line. These may be used also as bilge pumps. A complete system of bilge suction piping runs into each compartment, so that individual compartments may be pumped out in case of leakage.

The system of ventilating the various hospital wards on the vessel is interesting. An 80-inch steel plate blower, driven by an electric motor, direct attached, and capable of furnishing 8,000 cubic feet of air per minute, takes its supply of air through a ventilator leading from the hurricane deck down to an air-filtering chamber. The inlet to the blower is connected to the outlet of this filtering chamber. After filtration the air is forced through a system of brine-cooled coils, which lowers its temperature to about 40 degrees. Precipitation of moisture in the air occurs, the cool, dry air passing continuously from the cooling chamber to the air-heating chamber. This chamber is fitted with a number of steam coils—the air passing on the outside of the coils, and having its temperature raised to about 70 degrees F. The air then passes on through a system of ducts to the various wards. The air inlets to the wards are well distributed near the ceiling, and are controlled by dampers operated by thermostats, which are entirely automatic in their operation, and exceedingly sensitive and accurate. The temperature in the wards may be controlled within 1 degree F. This perfection of regulation is dependent upon the condition required by the board of physicians, viz.: that all windows and doors in the various wards shall be kept closed.

Near the floor line the vitiated air is removed through another series of ducts, this being done by an 80-inch steel plate motor-driven exhaust fan, having its in-take connected thereto, and delivering the vitiated air into the space between the inner and outer smokestacks. The air is completely renewed in the various wards every two and one-half minutes. When it is realized that approximately 11,500,000 cubic feet of air is treated in the manner described every 24 hours, some idea of the magnitude and importance of the work is obtained. Two independent exhaust fans remove the foul air from the laundry and the morgue, and deliver it to the space between the smokestacks fresh air being taken in through suitable ventilators.

There have been installed for operation in 1908 two separate refrigerating equipments—one machine having an air cooling or refrigerating capacity equal to the melting of 65 tons of ice in 24 hours. This machine will be used for the purpose of cooling and dehumidifying the air to be introduced into the hospital wards in such a manner as to reduce its temperature to 40 degrees F., and give it a relative humidity of 50 percent, at a time when the outside temperature may range from 80 to 90 degrees F.; after which this air is reheated to 70 degrees F. by means of steam coils. This system will undoubtedly be the direct means of saving the lives of many children during the hot and sultry days and nights. It may



MIDSHIP SECTION, SHOWING SCANTLINGS OF HULL.

be here noted that 1 ton of refrigeration is equal to the absorption of approximately 284,400 British thermal units (318,500 units per ton of 2,240 pounds). The other machine has a capacity of 5 tons in 24 hours, figured on the same basis. This machine will be used for the purpose of cooling the various refrigerators now existing on the vessel, and for the cooling coils of the circulating system for drinking water, as well as for making 400 pounds of ice daily.

The 65-ton compressor is of the horizontal double-acting

type, with steam and ammonia cylinders arranged in tandem. The ammonia cylinder is 14 inches diameter by 24 inches stroke, and the steam cylinder 20 inches diameter by 24 inches stroke. The engine is fitted with Corliss valve gear and a fly-wheel. The condenser is of the counter-current inner-tube type, consisting of 1½-inch water pipe and 2-inch ammonia pipe, the ammonia passing through the annular space formed between the two pipes. All screw threads have sweated and flanged joints made up with lead gaskets. The ammonia receiver is of wrought iron 16 inches diameter by 8 feet long, and tested to a pressure of 1,000 pounds per square inch. The brine coolers are of the counter-current inner-tube type, consisting of 2-inch brine piping inclosed in 3-inch ammonia pipe; ammonia being expanded in the annular space between the pipes. The coolers are erected in sections, and provided with valves, so that any section may be cut off without interference with any other section.

The 5-ton compressor is of the vertical, inclosed, twin-cylinder type, mounted on a bedplate and direct connected to a vertical engine. The compressor cylinders are 5 inches diameter by 9 inches stroke. The engine is 8 inches diameter by 8 inches stroke. The condenser, the ammonia receiver and the brine cooler are of the same general construction as that provided for the 65-ton equipment. The exhaust steam from both ammonia compressor plants is condensed by an independent surface condenser, with its own independent air and circulating pumps.

The propelling machinery of the *Boston Floating Hospital* consists of twin-screw vertical compound engines, 10 and 18½ inches by 14 inches stroke, indicating about 200 horsepower, and fitted with Joy valve gear, cutting off at about 70 percent of the stroke. Surface condenser, a combined air and circulating pump, measuring 10 by 10 inches and 12 by 12 inches, duplicate feed pumps, sanitary pump and fire and bilge pump are fitted. Owing to lack of funds, this machinery was not installed until Mrs. L. G. Burnham presented her yacht *Pilgrim* to the floating hospital authorities. The twin-screw machinery, condenser, air and circulating pumps were quickly transferred from the *Pilgrim* to the *Boston Floating Hospital*.

where they now do excellent duty. There are two four-bladed cast-iron propellers, 5 feet 6 inches in diameter, and operated at 140 revolutions per minute.

There are two Scotch boilers, 8 feet 9 inches diameter by 9 feet long, each containing two Morison corrugated furnaces, 33 inches inside diameter, and having a combined grate surface of 45 square feet, and 1,695 square feet of heating surface (ratio 37.7 to 1). The boilers furnish steam to the main engines, electric generator, two ice machines and auxiliaries, steam windlass, steam steerer and steam heating plant, and operate at 105 pounds per square inch. The bunker capacity is 20 tons. Under full power the vessel makes about 7 knots.

The electric generator is of General Electric manufacture, direct connected, 20 kilowatts, at 110 volts and 220 amperes, with one steam cylinder 9 inches diameter by 7 inches stroke, and runs at 360 revolutions per minute. This generator furnishes light for the whole vessel and gives power to the motors for driving the pressure and exhaust blowers which furnish fresh air to wards, as well as exhausting the foul air from them, and from the laundry and morgue.

During the year 1906 about 500 permanent patients and 800 day patients were treated. The list of diseases treated in each class, giving some idea of the work being done, shows 79 distinct ailments for the permanent patients and 75 for the day patients. Omitting duplications, no less than 129 diseases were handled. The cost of the vessel, complete, was about \$150,000 (£31,000).

THE MOST SUCCESSFUL DIMENSIONS OF STEAMSHIPS IN RELATION TO ECONOMY.

BY OTTO ALT, DIPLOM-INGENIEUR.

The question of the most efficient building of a technical structure—of a steamer in the present case—is to be considered the fundamental problem of technical science. It encroaches upon several provinces of human knowledge, and it may be solved only by the laws of physical science and of the science of economy. Without giving a comprehensive examination, I might give at the start as a criterion for a maximum of profit: The values produced by the object must be a maximum; the values consumed or destroyed by the object must be a minimum.

But this formula, seemingly so simple, may be perfectly realized only when all co-operating factors permit an exact definition. The problem will remain unsolved while there are actions of technical or economical nature, the laws of which are unknown. New laws are discovered with every progress of development, and their numerical value will be fixed in due course. So we will be only gradually successful in reaching the desired goal.

Starting from the above basis the question is: what ship is the most profitable in a given condition of trade, and how was this result obtained? In answering this question it is necessary only to take the merchant's position, and to look at the profit and loss account, either of a joint stock company or of a private undertaking. This account for the single object might be of the following form:

Credits.

Income from freightages, forwarding of passengers, etc.—*B*

Debits.

Operating expenses.....*D*

Insurance premiums, bills of exchange, maintenance money, costs of stationary and necessary incidents, costs for repairs.....*b₁*

Costs for fuel for the ship's power.....*b₂*

Interest.....*c*

Depreciation.....*a*

Net gain.....*E*

Supposing the capital agrees with the book value of the ship in the first year, it results that

$$A \text{ (inventory of ship with machinery)} = C \text{ (capital stock)}, \quad (1)$$

and, further, at the end of the first year:

$$E = B - D - (a + c). \quad (2)$$

The gain will be found in a similar way for other years, so that results for the whole length of the ship's life will be

$$\Sigma E = \Sigma [B - D - (a + c)]. \quad (3)$$

We might call this the equation of profit. The discussion of this equation (the complete connection of which we shall show later) naturally is simple: — ΣE will be a maximum, if ΣB is as high as possible, and ΣD , Σa and Σc are as low as possible.

In order to find out this maximum the laws must be known to which these quantities are subjected. All the single terms of equation (3) are not independent of one another, and to become perfectly aware of their relations a further analysis has to be made.

The value ΣB , i. e., the income from freightages, forwarding of passengers, etc., for ships undertaking regularly the same route (and for simplicity we speak here only of these), is proportional to the carriage rate i per ton of cargo (under which is to be understood quite generally freight and passengers), to the carrying capacity P and to the number of voyages n ; taking for the point of departure the unit of time, one year = 365 days. The number of voyages n is then:

$$n = \frac{365 - q}{2 \times e} = 12 V \left(\frac{365 - q}{e} \right) \quad (4)$$

whereby q means the number of days per year lying in harbor, in dock, etc.; e is the distance between the assumed harbors in nautical miles, and V signifies the speed of the ship in knots.

The result of these considerations is:

$$\Sigma B = \Sigma 12 V \left(\frac{365 - q}{e} \right) \times i \times P \quad (4)$$

In case the known quantities q , e , i are constant, as may be supposed for simplicity—but this is not at all necessary—the equation will become

$$\Sigma B = \Sigma \frac{k_1 \times V \times P}{12 (365 - q) i} \quad (5)$$

where $k_1 = \frac{e}{12 (365 - q) i}$

The tonnage P is further defined by the following equation:

$$P = W - (W_1 + W_2 + W_3 + W_4), \quad (6)$$

where W = displacement,

W_1 = weight of structure,

W_2 = weight of machinery,

W_3 = weight of fuel,

W_4 = weight of the fittings, stores, equipment, etc.

If for brevity $W_1 + W_2 + W_3 + W_4$ is accepted to be = ΣW_a , we have, from (5) and (6):

$$\Sigma B = \Sigma [k_1 \times V (W - \Sigma W_a)]. \quad (7)$$

As the profit and loss account will show, D is made up of Σb_1 and Σb_2 . John Inglis has shown¹ that b_1 , the costs of repairs, salaries, insurance, etc., is proportional to the gross tonnage, i. e., nearly independent of all other variations. To prevent too great complication, these expenses may be considered proportional to the value of the cargo, consequently

$$b_1 = m \times i \times P,$$

or, as it may be written: in case $k_2 = m \times i$,

$$b_1 = k_2 \times P. \quad (8)$$

¹ Transactions Institution Engineers and Shipbuilders in Scotland, 1895-1896, page 187.

The value of b_2 , the cost of fuel, is proportional to the number of voyages n , to the cost of 1 ton of fuel i' and to the weight W_3 of fuel consumed in one voyage, or, using the abbreviation:

$$k_3 = \frac{12 (365 - q) i'}{e}$$

$$b_2 = k_3 \times V \times W_3. \quad (9)$$

The debits, therefore, are, from equations (8) and (9),

$$\Sigma D = \Sigma (k_2 \times P + k_3 \times V \times W_3). \quad (10)$$

The sum to be written off, a , is proportional to the ship's building value A , and inversely proportional to the ship's length of life t ; therefore,

$$a = \frac{A}{t}$$

Supposing t to be constant, $i, e,$ to be not influenced by variations in the weight of the ship's hull, etc., then

$$a = k_4 \times A, \text{ and } k_4 = \text{constant}. \quad (11)$$

Up to now only those values have been considered which are of interest to the shipowner. But the ship designer has an interest, too, in the building value A . It is possible to state the interdependence of A upon other values in the following way: the building value of the structure is composed of the building value of the ship's hull, of the machinery and of the equipment. Suppose the cost per ton of the ship's hull = i_1 , those per ton of the machinery = i_2 and those per ton of fittings = i_4 , then

$$A = i_1 W_1 + i_2 W_2 + i_4 W_4, \quad (12)$$

where it may be noted that i_1, i_2, i_4 may be different in different alterations of the values W_1, W_2 and W_4 .

These results change (11) and (12) into

$$\Sigma a = \Sigma k_4 [i_1 W_1 + i_2 W_2 + i_4 W_4]. \quad (13)$$

The interest on the capital, c , is quickly disposed of, if supposed to be proportional to the building value (as before C is supposed to equal A); then

$$c = k_5 \times A,$$

or

$$\Sigma c = \Sigma k_5 [i_1 W_1 + i_2 W_2 + i_4 W_4]. \quad (14)$$

After having introduced for the single values characteristic values, these must be put into equation (3) in order to get a further idea of the problem. In a simple way we get:

$$\Sigma E = V (C_1 P - C_3 W_3) - C_2 P - C_4 (i_1 W_1 + i_2 W_2 + i_4 W_4), \quad (15)$$

in which C_1 has been written for Σk_1 , C_2 for Σk_2 , C_3 for Σk_3 , and C_4 for $\Sigma (k_4 + k_5)$.

The total gain, equation (15), being determinative, shows a dependence on the values: cargo capacity P , weight of the ship's hull W_1 , weight of machinery W_2 , weight of fuel W_3 , weight of equipment and of stores W_4 , speed of the ship V and the unit prices i_1, i_2, i_4 , as well as on the constants C_1, C_2, C_3 and C_4 .

If this value ΣE is to become a maximum, then, mathematically speaking, this relation must exist:

$$\delta (\Sigma E) = 0. \quad (16)$$

It would be superfluous to give the proof of the existence of such a solution, which is declared only formally by the relation (16).

The further question: how may the maximum of ΣE be surely found? consists of constructing the variations which these values may undergo, upon the basis of the laws of physical science and of the science of economy. One would start by trying to bring these variations into a form in which he might formulate the dependence of the values P, W_1, W_2 , etc., upon a system of variables, to be considered independ-

ently. We might call these variables the *fundamental variables*.

When we have to do with materials existing in nature—either the object is manufactured of them or it comes into contact with them during its life—certain other values enter into the problem, which are constant with a determined arrangement of materials (to use a very general expression). At every change in the arrangement, such as the transition of webs to deep frames, or of a spar-deck type to a trunk-deck type, etc., the constants will have other values; but they will keep these values for all variations of the fundamental variables. These latter values may then be considered to be the *fundamental constants*. The problem may then be considered solved if it is possible to express all influences by a certain group of variables and constants.

After having sketched quite generally the method, we must come to an agreement as to which values are to be considered fundamental variables and which are constants.

A ship's physiognomy is determined to a certain degree by its geometrical form, and this form is strongly influenced by the dimensions L, B, D , to which may be added the draft d , and the coefficients of displacement δ , of load waterline α , and of midship section β . These may be considered the fundamental variables. If it is possible to prove that all influences in respect to the expression ΣE are to be rendered by these values, the group of fundamental variables does not need any supplement. As to the fundamental constants, all that is necessary will be said later on. When the variables, as well as the constants (in case alterations are undertaken in the structure), run through distinct systems of value—such that it is possible to stay in the neighborhood of the maximum ΣE —then the values $W_1, W_2, W_3, \dots$ etc., by their introduction into the relation (15), enable the desired maximum of profit to be reached.

I suppose that the problem has neither been solved as yet in the generality, as given here, nor has it been formed into equations. At least, I did not find any case of this kind, in spite of exhaustive examination of the literature on the subject. Many authors, as Hamilton² and Inglis, confine themselves—using more empirical methods—to the inquiry into the influence of W and V (and therefore W_3 , too) on E only. Others think the maximum of ΣE to be identical with the

$$\text{maximum of } \frac{\text{useful displacement } P}{\text{displacement } W} = \frac{P}{W}. \text{ So William}$$

Froude³ says: "It seems a mere truism to state that the best ship for the performance of a given duty is that in which the useful displacement bears the largest proportion to the whole displacement." Examining this in connection with equation (15), and dividing both sides of the equation by W , there results:

$$\frac{\Sigma E}{W} = V \times \frac{C_1 P - C_3 W_3}{W} - \frac{C_2 P}{W} - \frac{C_4 (i_1 W_1 + i_2 W_2 + i_4 W_4)}{W} \quad (17)$$

and it is seen that this is not the case by any means.

Rather it might be supposed that the maximum value of

$$\Sigma E \text{ is identical with the maximum of } \frac{P - W_3}{W}, \text{ when } C_1$$

and C_3 are equal, and when the rest of the right side in equation (17) remains nearly constant. But Froude might

² Transactions of Institution of Naval Architects, 1883, page 256.

³ Transactions of Institution of Naval Architects, 1874, page 148; see also A. C. Holms, *Practical Shipbuilding*, London, 1904, page 2.

not have thought of that, for he says: "The useful displacement is regarded as that displacement which remains, after deducting the dead weights of the hull as completed, with proper regard to structural strength, and of the engine power necessary to drive the ship at the required speed, and of the coal which will be consumed on the voyage." Surely Froude's "useful displacement" is very important in the question of profit, but a laying of stress upon these values will cause wrong conclusions, as is shown by the relations (15) and (17).

It is thus to be seen how the nature of the problem of profit is reflected much more generally by the examination here undertaken; which besides is not bound to the recapitulation of the variations of W_1 , W_2 , etc., with the fundamental variables by a mathematical-analytical process; on the contrary, the proportions of dependence can be given graphically, anyhow.

In this general composition, it is to be supposed that already measures have been taken on the shipowner's part to determine the value of P or of the number of passengers, and, besides, of the speed, upon the basis of economical conditions. These values are dependent on a whole succession of special factors, on the special development of the shipowner's interests, on commercial intercourse, on the speed of production and of consumption, on the special route the ship is to go, etc. But even then considerable variations are possible, and, above all, those which are dependent on the engineer.

Under these limitations it is possible at least to declare the dependence on the dimensions which may be shown for the single values W_1 , W_2 , etc., which figure prominently in equation (15), or—in case they are known already—to introduce them here in the right way.

The analysis of the value—weight of structure—will cause considerable difficulty; remembering the following influences: The form of the hull under the waterline, above the waterline (freeboard), stability (the metacentric height $G M$), choice of dimensions, the stresses in longitudinal and transverse dimensions, local strains, racking strains, the choice of the general principle of construction—high class—spar deck, well deck, turret-deck vessel, web frames, hold beams, choice of propelling machinery: steam engine, steam turbine, etc. A discussion of all these influences would go too far; it is required only to show the integral parts. In order to simplify the problem, to separate and to clear the influences, we divide the weight of structure—corresponding to the character of the single stresses—into the weight of the longitudinal structure = w_1 , the weight of the transverse structure = w_2 , the weight of structure required for the local stresses = w_3 , and the weight of structure required by the racking stresses = w_4 .

Therefore, the relation exists:

$$W_1 = w_1 + w_2 + w_3 + w_4. \quad (18)$$

In order to find out the weight of the longitudinal structure—including all the uninterrupted, continuous parts of the structure—we consider the longitudinal bending moment. By Vivet⁴ and Alexander⁵ are laid down exhaustive examinations for the standard conditions: the ship is resting on a wave, with the length of the wave = L and with the height of the

wave = $\frac{1}{20} L$, referring to the dependence of the longitudinal

bending moment on the dimensions. Instead of the quite analytical formula given by Vivet we fall back upon the graphic data of Alexander, who, starting from the formula

$$M = \frac{W L}{\lambda}, \quad (19)$$

has shown the dependence of the bending moment on the

remaining dimensions, in considering the value λ , to be dependent on them.

Supplying the ship's place with a box (Fig. 1), as Froude, Normand⁶, Vivet and others did, the moment of resistance is

$$r = \frac{s \times D (3B + D)}{3} \quad (20)$$

in which terms the greater powers of s are neglected, and it results from the relation of weight that

$$w_1 = a_0 (B + D) \times s \times L, \quad (21)$$

and from the formula of the bending moment of a straight beam:

$$p = \frac{M}{r}, \quad (22)$$

the following dependence for w_1 :

$$w_1 = A_0 \times \frac{L}{D} \times \frac{B + D}{3B + D} \times M, \quad (23)$$

or

$$w_1 = A_0 \times \frac{L^2}{D} \times \frac{B + D}{3B + D} \times \frac{W}{\lambda}. \quad (24)$$

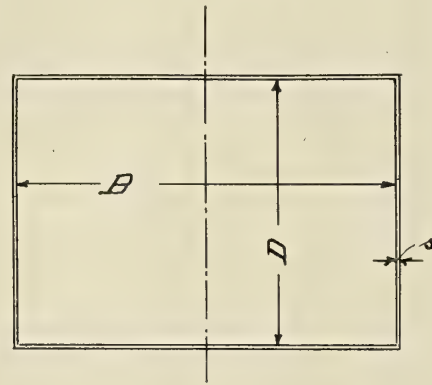


FIG. 1.

This formula is supposed to be employed in the following way: A study of strength for a choice of dimensions and of cross section of girder is to be made, and the weight of the longitudinal structure is found. According to equation (24) the constant A_0 may be calculated, which has the character of a fundamental constant. The variation of w_1 is then to be undertaken according to equation (24) for further examination. An analysis showed that the mistakes thus made are unimportant, in case L/D does not get too low; and that happened only because the hull was to be made stronger, on account of the local stresses, than the longitudinal strength required.

It has often been affirmed that the influence of δ is another important item for a ship at sea (namely, taking into consideration the forces of inertia), as it is stated by the formula of bending, enlarged by Alexander; as, for instance, Read;⁷ but these examinations are yet too unsystematic and unrelated to permit a conclusion.

The weight of the transverse structure, w_2 , does not permit such simple analysis; above all, because a great deal of very many different structural arrangements belong to it, the weight of which varies according to quite different laws. There are to be included in the transverse structure the frames, floors, web frames, hold beams, side stringers (in case these are fitted to assist the transverse structure), the stiffeners of longitudinal and transverse bulkheads, beams, etc.

As yet nothing is known about the dependence of the transverse structure on the dimensions (fundamental variables).

⁴ Bulletin de l'Association Technique Maritime, 1894, page 114.

⁵ Transactions Institution Naval Architects, 1905, page 116.

⁶ Bulletin de l'Association Technique Maritime, 1892, page 2.

⁷ Transactions Institution Naval Architects, 1890, page 179.

The examinations undertaken by Bruhn,⁸ who treated the problem of transverse strength according to the theories of the Italian engineer Castigliano, do not even show how the transverse bending moment and the fundamental variables act and react upon each other.

To take a general case, we imagine a section of a steamer with several decks, of the length l , cut off in the vicinity of the midship section (see Fig. 2). Then different exterior forces—pressure on the bottom and sides by cargo (or the weight of parts of the machinery), hydrostatic pressure, the weight of the structure present in the respective cross section—and some other forces besides will act upon it. The principal system,⁹ statically determined, will be formed then by that part of the cross section which remains after the beams (the so-called supernumerary struts) have been removed. To keep the former strength, instead of the removed beams, some structural items (see Fig. 3) are to be fixed, which introduce forces, or forces and moments, if a solid fastening exists, by strong beam knees. (See Fig. 4; both figures are taken from a tank steamer.)

First considering the moment of the exterior forces for every point of the principal system, it is necessary to state what are the conditions for the development of the greatest bending moment. But it is impossible to give general statements, authentic for all cases and types of ships, etc. It depends on the character of the cargo, on the situation and number of the longitudinal bulkheads, on the ship's condition

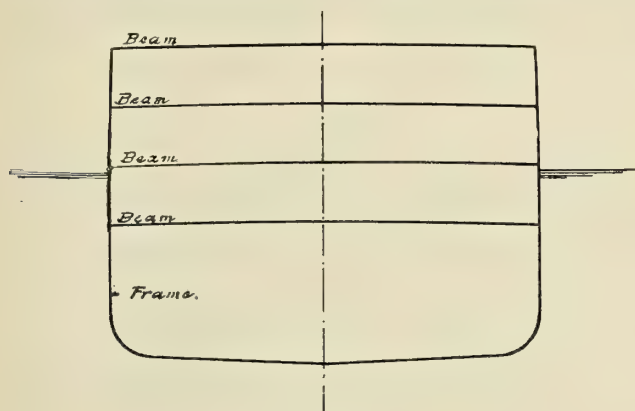


FIG. 2.

—if in dock, in calm or in rough water (cf. Bruhn). Here one case may be taken arbitrarily, and the bending moment may be derived for it; for instance, a steamer carrying oil in bulk, with a cross section like Fig. 5. The cross section may be filled with water ballast; and the draft—according to the situation of the cross section in the trough of the sea—may be h' . The moment acting on the floor in the point B is about

$$M = \frac{\gamma \times l}{8} (\beta_1 v_2 \times B^2 \times h + b^2 \times h_0) + \frac{\gamma \times l}{6} [h^2 (3h_0 + h) - (h')^2] - \beta_2 \times v_2 \times \frac{\gamma \times l}{8} \times B^2 \times h' - \frac{S \times B}{2} \quad (25)$$

S is to be supposed to equal the third part of the difference between weight and displacement:

$$S = \frac{\gamma \times l}{3} [(\beta_1 \times B \times h + b \times h_0) - \beta_2 \times B \times h'] \quad (26)$$

⁸ Transactions Institution Naval Architects, 1902, page 270, and 1904, page 193.

⁹ Compare the competent treatises on bridge building.

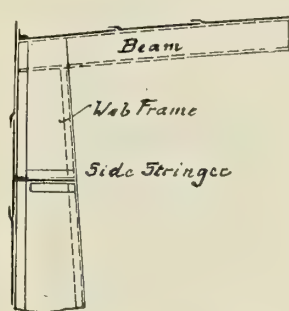


FIG. 3.

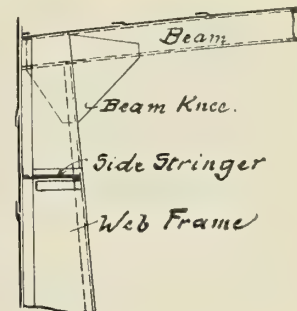


FIG. 4.

Besides the values already explained, in Fig. 5, in (25) and (26):

γ means the specific gravity of water.

β_1, β_2 mean the coefficients of midship section up to the main deck and to the draft h' .

v_1 and v_2 are coefficients, being nearly unity (and may be calculated exactly).

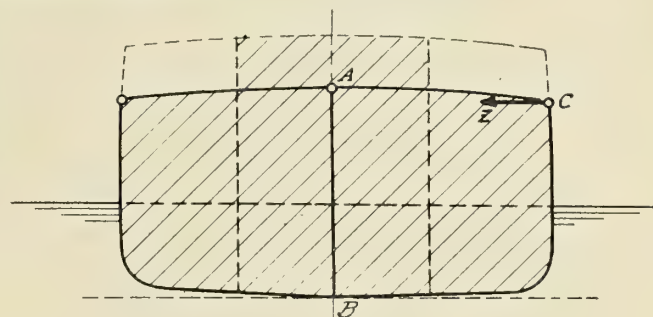


FIG. 5.

At a standard wave, for sagging condition, $h' = d - \frac{L}{32}$.

in case d is the original draft. But it is easily proved that this value does not characterize the maximum bending moment at all; on the contrary, it agrees with the value of h' , which is to be calculated by the relation

$$\frac{\delta M}{\delta h'} = 0 \quad (27)$$

It results from equations (25) to (27), at all events, that a dependence may be derived from the dimensions.

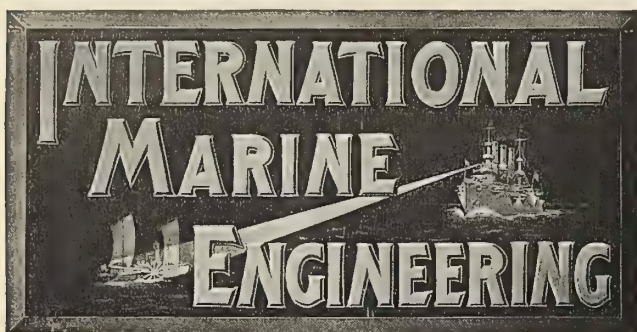
(To be Concluded.)

American Warship Construction.

The decided falling off in warship construction in the United States during the past four years is clearly shown in the following table, which gives in each instance the vessels under construction on July 1:

	1904.	1905.	1906.	1907.	1908.
Battleships	13	13	9	5	4
Armored cruisers.....	11	9	8	3	1
Protected cruisers.....	2	..	3	3	1
Gunboats	2	2	..	..	..
Torpedo craft.....	5	2	..	..	5
Submarines	..	..	4	4	7

The displacement under construction, excluding the submarines, has fallen off from 344,859 tons in 1904 to 313,845 tons in 1905, 251,906 tons in 1906, 127,930 tons in 1907, and 93,750 tons in 1908.



Published Monthly at

17 Battery Place New York

By MARINE ENGINEERING, INCORPORATED

H. L. ALDRICH, President and Treasurer

GEORGE SLATE, Vice-President

E. L. SUMNER, Secretary
and at

Christopher St., Finsbury Square, London, E. C.

E. J. P. BENN, Director and Publisher

SIDNEY GRAVES KOON, Editor

Branch { Philadelphia, Machinery Dept., The Bourse, S. W. ANNESS.
Offices { Boston, 170 Summer St., S. I. CARPENTER.

Entered at New York Post Office as second-class matter.

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the month.

Speed Trials of the Lusitania.

One of the papers before the Institution of Naval Architects in April is being republished this month (concluded from July) in the shape of a discussion of the speed trials and service performance of the Cunard steamer *Lusitania*. This is particularly apropos at the present moment, when both the *Lusitania* and the *Mauretania* are making new records on almost every trip.

In the latter half of May, the *Lusitania* ran across the Atlantic by what is known as the long route, in four days 20 hours and 22 minutes, the average speed being 24.83 knots. During the four full days of this trip the smallest daily run was 622 nautical miles, while the greatest, 632 miles, was a record, beating by 5 miles the previous record day's run, made by the same ship. A trip during the last week of May by the *Mauretania*, in which one propeller of the four was useless, showed an even more remarkable result in that the average speed, thus crippled, was no less than 24.64

knots. These runs have, it is reported, made the ships eligible for the British government subsidy, which was dependent upon a sustained round-trip speed of 24½ knots.

In the first half of June, the *Lusitania* made another record-breaking trip, during which the long route of 2,890 nautical miles was covered in four days 20 hours and 8 minutes, the average speed for the entire trip having been 24.88 knots. The best day's run during this trip was 641 nautical miles, the actual steaming period having been 25 hours 16 minutes (this variation from the 24 hours in a "land" day having been caused by the difference in longitude between the points of departure and conclusion of the day's run). The speed for this day figured out at 25.37 knots.

On her next westward trip, ending July 10, the *Lusitania* covered 2,891 miles in four days 19 hours and 36 minutes, the average speed having been 25.01 knots. One day's run of 643 nautical miles accounts for a speed during that day of 25.45 knots.

These ships are exciting such intense interest in all quarters by reason of their great size, superior appointments and especially of their high speed, that the paper before the Institution will be read with much interest wherever it is seen.

The Brazilian Battleships.

Comment has been rife during the past few weeks regarding the three Brazilian battleships under construction in England by the Elswick Works and Vickers, Sons & Maxim, Ltd. It has been stated, and vociferously denied, that the ultimate destination of these ships was Tokyo, and that the probabilities were strong against their ever flying the green and gold emblem of Brazil. Such a sale of ships under construction would be by no means novel; for in 1897 Brazil sold two cruisers to the United States; in 1903 Argentina sold two armored cruisers to Japan; in 1895 Japan bought from Chile, through Ecuador, the first protected cruiser ever built; in 1903 Britain purchased from Chile two battleships then nearly completed; and it is said that within the past few weeks Brazil has actually sold to Japan one of the torpedo boats forming part of the program of twenty-seven ships ordered more than a year ago.

Interest naturally centers, not only in the ships themselves, but in the sudden disturbance of the balance of naval power which would result should these vessels be transferred to Japan after having been almost completed for another nation. It may be remarked that the battleship strength of the Japanese navy, including ships built and building, covers eighteen vessels, with an aggregate broadside fire of 84,920 pounds (nothing under 4-inch guns being counted). The three Brazilian battleships have between them a broadside fire of 26,820 pounds, which would represent an increase to the Japanese broadside of 31.6 percent.

Japanese interests are wholly in the Pacific, and here they brush up against no one more strongly or more insistently than against the United States. The present fleet of sixteen American battleships now traversing the Pacific on a cruise, which has excited world-wide attention, has an aggregate broadside fire of 87,010 pounds, or only slightly greater than the present Japanese figures. The disturbance of the balance of power in the Pacific by the sudden addition of these three ships to the Japanese navy would thus be very startling. It is true that the American navy includes thirteen other battleships, with an aggregate broadside fire of 79,535 pounds, but these are not on the spot. Four of them are totally uncompleted and three others are undergoing extensive refits.

Regarding the ships themselves, considerable secrecy has been maintained, and such reports as have come out differ materially on important points. From best information, however, each ship appears to be of 19,250 tons and 21 knots. The main battery is given as twelve 12-inch rifles, mounted in pairs in six turrets, four turrets being on the center line and one on either beam; this gives ten guns in broadside. The secondary battery is given as twenty-two 4.7-inch guns. The aggregate broadside fire figures out at 8,940 pounds. This compares with 6,800 pounds for the *Dreadnought*, 7,234 pounds for the *St. Vincent*, 7,000 pounds for the *Satsuma*, 6,790 pounds for the *Danton*, 9,120 pounds for the *Delaware*, and 9,912 pounds for the *Nassau*.

It is thus seen that, with the exception of the new American and German ships, the Brazilians are totally without a close competitor with regard to broadside fire. Their acquisition by any power would mean an extremely effective addition to that power's first line of defense, and the outcome will be awaited with much interest.

The Combination of Engines and Turbines.

For a long time it has been recognized that the steam turbine could use expansion ratios far beyond the limits of economical operation in piston engines. This does not mean that the latter cannot be made to expand the steam down to as low a pressure as may be reached by the turbine; but it does mean that were such an engine designed and built, the size and weight of the low-pressure cylinder would be so excessive as to preclude all possibility of its being put commercially into satisfactory use. Not only this, but the immense surface thus formed would, unless extraordinary precautions were taken, give rise to a very great amount of cylinder condensation—the one source of greatest loss in the operation of the piston engine. By proper arrangement, however, of the turbine-drum diameter, and the passages between the blades, accommodation can be made to fit a suitable speed of rotation to the natural velocity of flow of the steam at these low pres-

ures, and advantage taken of much of the great energy available from the steam in expanding through the last two or three inches of vacuum.

This peculiarity of the turbine was perhaps never more strikingly illustrated than on the westward passage of the steamship *Mauretania* at the end of May. As stated in another column, the ship made on this trip a new record of 635 nautical miles in one steaming day, and maintained an average speed over a course of 2,889 miles of 24.64 knots. This achievement was widely heralded as a great marvel, due to the fact that it was made with three, instead of the four, propellers with which the ship is ordinarily driven, one of the high-pressure propellers being temporarily out of commission. The steam, of course, was passed through simply one high-pressure turbine, and from this one to the two low-pressure turbines. The expansion was carried out to a very high degree, and such is the remarkable adaptability of the turbine to balance itself against whatever work may be placed upon it, that the total power developed in the three turbines, using approximately the same amount of steam as had previously been used in four, could not have been far different from the maximum ordinarily obtained with all four engines under operation. This fact, involving the utilization of the last few pounds of pressure above a vacuum, accounts theoretically for what, under any circumstances, must be considered a remarkable performance.

The last of the papers which we shall publish, read in April before the Institution of Naval Architects, appears in this number; and is concerned with the adoption of steam turbines for the utilization of these low pressures, after the steam has been previously passed through high-pressure engines of the reciprocating type. The main advantage of the combination lies in the ready adaptability of the piston engine to maneuvering, and more particularly to reversing.

One important point in this connection is the fact that turbines may be fitted to take up their burden at a pressure as low as that at which a piston engine is ready to discard the steam. This would involve no increase in the consumption of fuel; the only charge against it would be that of carrying the extra weight of the turbine and its accessories. If to offset this, while maintaining the same ship speed as before, we decrease the size of both engine and turbine, making the combination equal in power to the former piston engine alone, we may thereby decrease the coal consumption as compared with previous figures and propel our ship more economically from this point of view than before. The whole question is a commercial one, consisting of the balancing of improved speed or coal consumption, or both, against increased cost of installation and consequently increased fixed charges. It is a very interesting proposition, and the two or three important installations at present under way will be closely watched by engineers.

Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, reports the following percentages of completion of vessels for the United States navy:

				May 1.	June 1
BATTLESHIPS.					
	Tons.	Knots.			
South Carolina.....	16,000	18½	Wm. Cramp & Sons.....	45.9	49.
Michigan.....	16,000	18½	New York Shipbuilding Co.....	50.7	53.
Delaware.....	20,000	21	Newport News S. B. & D. D. Co	22.8	27.4
North Dakota.....	20,000	21	Fore River Shipbuilding Co....	31.6	35.7
ARMORED CRUISER.					
Montana.....	14,500	22	Newport News Co.....	98.	98.8
SCOUT CRUISER.					
Salem.....	3,750	24	Fore River Shipbuilding Co....	96.4	97.1
TORPEDO BOAT DESTROYERS.					
Number 17.....	700	28	Wm. Cramp & Sons.....	15.9	21.3
Number 18.....	700	28	Wm. Cramp & Sons.....	15.4	19.7
Number 19.....	700	28	New York Shipbuilding Co....	16.2	22.4
Number 20.....	700	28	Bath Iron Works.....	9.7	11.5
Number 21.....	700	28	Bath Iron Works.....	9.3	10.9
SUBMARINE TORPEDO BOATS.					
Cuttlefish.....	—	—	Fore River Shipbuilding Co....	99.	99.
Number 13.....	—	—	Fore River Shipbuilding Co....	40.5	45.3
Number 14.....	—	—	Fore River Shipbuilding Co....	39.9	45.2
Number 15.....	—	—	Fore River Shipbuilding Co....	34.	44.9
Number 16.....	—	—	Fore River Shipbuilding Co....	34.2	45.1
Number 17.....	—	—	Fore River Shipbuilding Co....	13.6	27.5
Number 18.....	—	—	Fore River Shipbuilding Co....	13.4	23.1
Number 19.....	—	—	Fore River Shipbuilding Co....	13.	23.1

International Yacht Race in Honor of Columbus.

Spain intends to commemorate the 3d of August next, the anniversary of the departure from Palos of the ships *Santa Maria*, *La Pinta* and *La Nina*, by yacht races, which, starting from that port, will make the first stage of the voyage which was made by Columbus. The races will be from Palos, whence Columbus set sail, to the Canaries. The distance is 700 miles, requiring several days' navigation.

This year, by a happy coincidence, high water is at 6 o'clock in the morning on the 3d of August (this being the day and hour that, 415 years ago, the *Santa Maria*, *Pinta* and *Nina* set sail at high water), which will be taken advantage of this anniversary by the American-Spanish yachts to perform the first part of the journey Columbus made in 1492. The Mayor of Puerto Palos has arranged that the first sailing yacht to arrive shall anchor at the Palos quay, bearing east, and that those arriving later shall anchor up-stream, in the order of their arrival; so that on the 3d of August all the vessels which shall take part in the regatta shall pass over the exact spot from which Columbus' caravels sailed.

ENGINEERING SPECIALTIES.

Lubrication of Marine Engines.

No other line of machinery demands more careful and scientific lubrication than that on board a modern steamship. Here troubles from overheated or scored bearings are scarcely excused under any circumstances, while at the same time there is demanded the highest degree of cleanliness, efficiency and economy. The lubrication of machinery is, in the first place, really a greasing proposition. It is grease that is needed between the surfaces of a journal and its bearings; and oil has never been anything else than a means for carrying the grease globules where they are needed. Why not, then, discard oil and use ordinary grease? Simply because ordinary grease contains substances which in time gum up the cup and bearing and cut the journal.

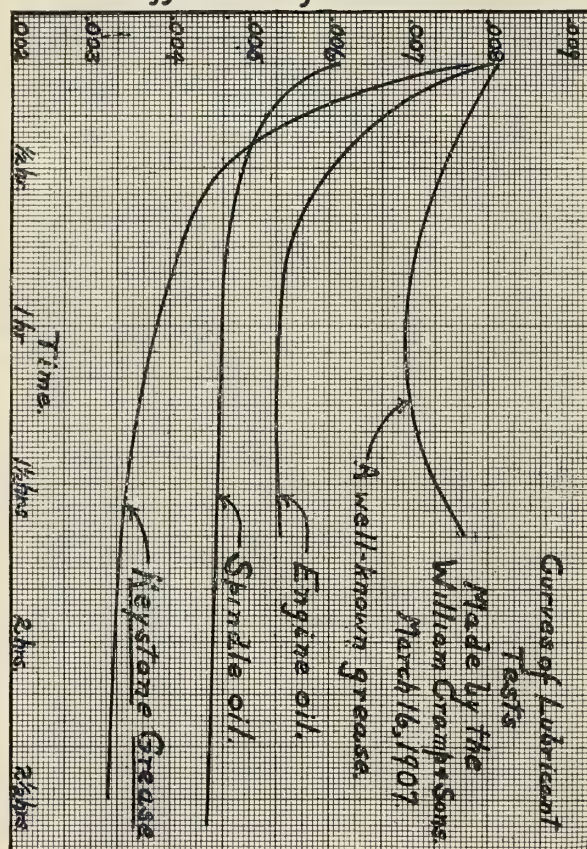
Keystone grease,* however, is said to be utterly and absolutely different from any other lubricating grease on the market. In the first place it is absolutely free from resin, resinous oils, talc, asbestos, beeswax, soapstone or any other of the adulterants commonly used in other greases to give

artificial body. It is a pure, clean petroleum product, containing nothing but grease globules in their most perfect form. It is a lubricating grease that needs no wasteful carrying agency, as is true of grease in oils, nor does it need help of any kind to enable it to maintain a greasing film, no matter what the temperature or pressure.

Being a purely mineral product this grease contains no stearine. Animal oils and greases always contain stearine; and when they are used for lubricating piston rods, valve stems, steam chests, or steam cylinders, a chemical action is inevitable, which produces stearic acid. This is very active, attacking iron readily, and even copper. A sample of Keystone grease analyzed by William Cramp & Sons' chemist was found to be entirely free from all deleterious substances, such as would cause decomposition of the grease or would have any detrimental effect upon metal surfaces.

A favorite method of applying greases is to place them in contact with the bearing, and depend upon the heat of friction to melt the grease and cause it to flow upon the bearing. In other words, the bearing must always be hot enough to melt the grease. Keystone grease does not change consistency under any variation of temperature up to 600 degrees F. This means that cold weather does not hinder its flow, nor does heat cause it to flow wastefully. It is made in eight different densities, ranging from a liquid suitable for fine high-speed service to a solid that will maintain a greasing film under the heaviest known pressures. These heavier grades are all applied by means of especially designed pressure cups,

Coefficient of Friction.



which are automatically or hand regulated as desired, and in either case the flow is uniform and certain. The fluid grades are applied very much as is ordinary heavy oil, but much less is required.

The lubricating efficiency of an oil or grease is indicated by the coefficient of friction obtained with its use upon a standard testing machine. The card shows the curves plotted from the tests of lubricants conducted by the Cramp Company. It will

* Keystone Lubricating Company, Philadelphia, Pa.

be seen that the curve plotted from the results of the test on Keystone grease falls much lower than that of any of the other lubricants tested. The average coefficients of friction of the different lubricants were as follows: "Well-known" grease, 0.007; engine oil, 0.0053; spindle oil, 0.0047; Keystone grease, 0.0034.

Oddesse Pumps.

Of the many distinguishing features of this pump, produced by the Oddesse Pump Company, 47 Victoria street, London, S. W., the most striking is the great simplicity of the valve gear. The pump has none of the links, levers, rockshafts and external wearing parts of the ordinary duplex pumps, the valves being actuated by means of a very simple patented device consisting of an inclined slipper engaging in an inclined slot in a block partaking of the same motion as the piston rod. No setting of the valves is required; they can be taken out and examined without the possibility of not being properly replaced, and there are no nuts requiring accurate adjustment or that can get loose. As there are no external wearing parts whatever, the whole of the lubrication can be most effectively carried out by means of a lubricator on the steam pipe.

Another point of particular interest in the Oddesse pump is the means employed for bringing the piston to rest at the end of the stroke. In all ordinary duplex pumps the piston is allowed to pass over the exhaust ports, thus cushioning the exhaust steam. This method however, has the great drawback of being incapable of alteration for various speeds and

and short stroking impossible, no matter what speed and against what pressure the pump may be running.

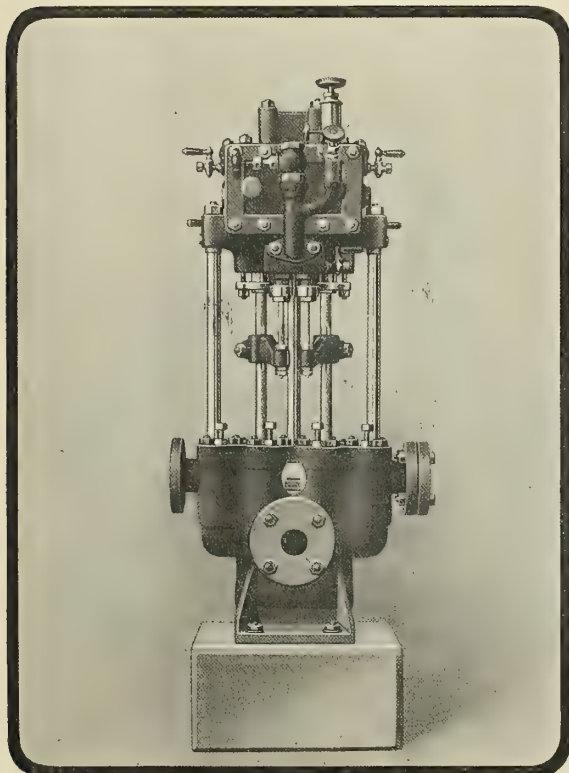
Owing to the simplicity of the valve motion, the expansion valves can be fitted very simply without the use of any levers, glands or joint pins, so that the Oddesse gear with cut-off valves remains far less complicated and more durable than the ordinary duplex gear without them. The point at which the steam is cut off is regulated by means of handles placed on either side of the valve chest, and in the larger sized pumps the expansion valves are automatically controlled by the slightest variations in the length of stroke made. These cut-off valves enable the pump to run at enormously high speeds without the slightest danger of any knocking of the piston or hammering of the pump valves, and it is this that makes this pump so serviceable on board ship as a ballast, bilge or fire pump.

Tantalum Lamps.

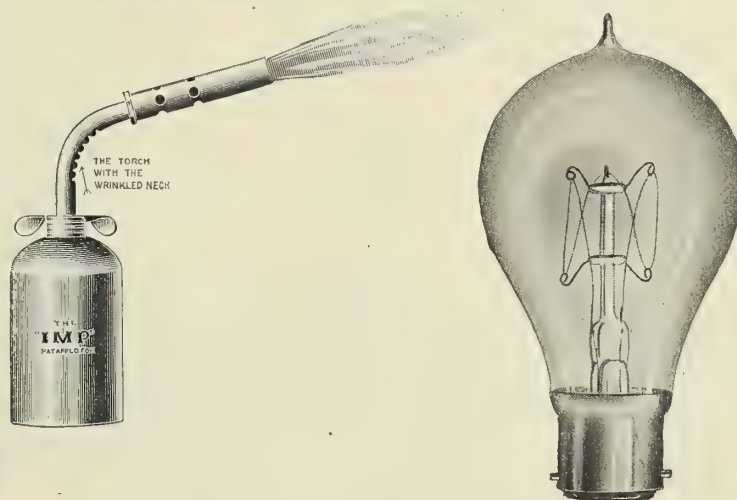
A new type of wire filament lamp has been placed on the market by Siemens Bros. Dynamo Works, Ltd., London, who are introducing a 25-volt tantalum lamp, for use with automatic transformers and on low voltage circuits generally. The lamps can be burned in any position.

The new lamp is for voltages of 24 or 25, and is supplied in 8 or 16 candlepower. Like all other types of tantalum lamps the filament is very strong; the 25-volt tantalum lamp may be used for train lighting and private house installations, especially in connection with small transformers. The bulb is smaller than that of the ordinary carbon filament lamp, but similar in shape.

These lamps have been supplied to all the ships of the Hamburg-American Line, and it is said that on the *Graf Waldersee* one dynamo is sufficient for the lighting, where three were previously used. Tantalum lamps are also in use on the *Lusitania* of the Cunard Line.



pressures. Hence the chief cause of short stroking at low speeds and knocking at high speeds. Again, when the pump is run at anything above moderate speeds, the compression of the exhaust steam at the end of the stroke causes a recoil of the reciprocating parts, which naturally results in a hammering action on the pump valves. In this pump the steam can be cut off at any point during the stroke, and the piston is brought quietly to rest by means of the fall of pressure in the cylinder, thus rendering a recoil of the reciprocating parts



A Powerful Gasoline Blow Torch.

The "Imp" torch is a patented device which, it is said, will do as much work as most of the larger torches, with the advantage of compactness, simplicity and cheapness. It is entirely automatic in operation, has no pump or valve, needs no tools, starts with a match and gives a perfectly clean, powerful Bunsen flame for over two hours on four ounces of gasoline. The corrugated neck increases the heating surface to such an extent that the flame of a match easily generates gas enough for starting, after which the mixing tube renders further attention unnecessary.

The "Imp" is designed for electricians, automobilists, the handy man and anyone who wants intense, clean heat, cheaply

and quickly. It is made by the Frank Mossberg Company, Attleboro, Mass.

Magnalium.

An alloy consisting mainly of aluminum, but with a small percentage of magnesium, has been developed in Germany, and, on account of some rather remarkable qualities, has been put to various uses. It is being handled in the United States by Morris R. Machol, 32 Park Place, New York.

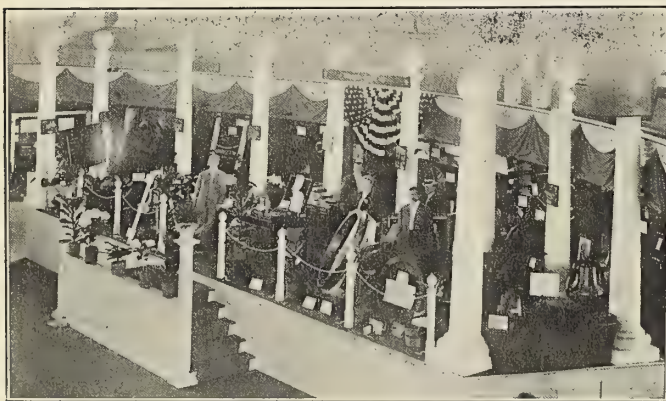
The alloy is very slightly lighter than aluminum, and may be used either cast, drawn or rolled. When cast in dry sand the tensile strength is said to be from 18,000 to 21,000 pounds per square inch, with a reduction in area of $3\frac{3}{4}$ percent. Cast in iron chills, the strength is about 4,000 pounds greater than in dry sand. A tensile strength of as much as 42,500 pounds is said to have been obtained by special treatment. Soft rolled sheets are accredited with a strength of 42,000 pounds and 15 percent reduction of area; hard rolled sheets with 52,000 pounds and 3 percent reduction of area. Wire drawn from this alloy is given 41,000 pounds and 10 percent reduction, while 53,000 pounds has been shown when the raw material was forged before drawing. When magnalium contains less than a certain percentage of aluminum it cannot be rolled, but can readily be drawn. In such case a bar has been shown to give 60,000 pounds per square inch tensile strength and a tube 74,000 pounds.

The metal is very close-grained, thus facilitating polishing. It can be turned at very high speed in a lathe, and will readily hold lacquer for polishing or etching. It is hard, but when annealed may be made so ductile that it can be rolled or beaten like silver. It is said to be practically impervious to the action of most acids and salt water, and as such to be peculiarly fitted for use in ship work and elsewhere, where it is necessary to resist corrosion.

Among the numerous articles which have been made of magnalium might be mentioned parts of delicate and other machinery; steam and water valves; optical instruments, and various parts of automobiles, bicycles and gasoline engines, such particularly as engine bases, etc.

Vanadium Steel.

Much interest was manifested in a recent exhibit by the Vanadium Sales Company, of Pittsburg, Pa., of vanadium products. The variety of the specimens was considerable, including open hearth and crucible steels treated with vanad-



ium, vanadium cast iron, vanadium steel castings and vanadium copper and bronze, of considerable hardness, strength and ductility.

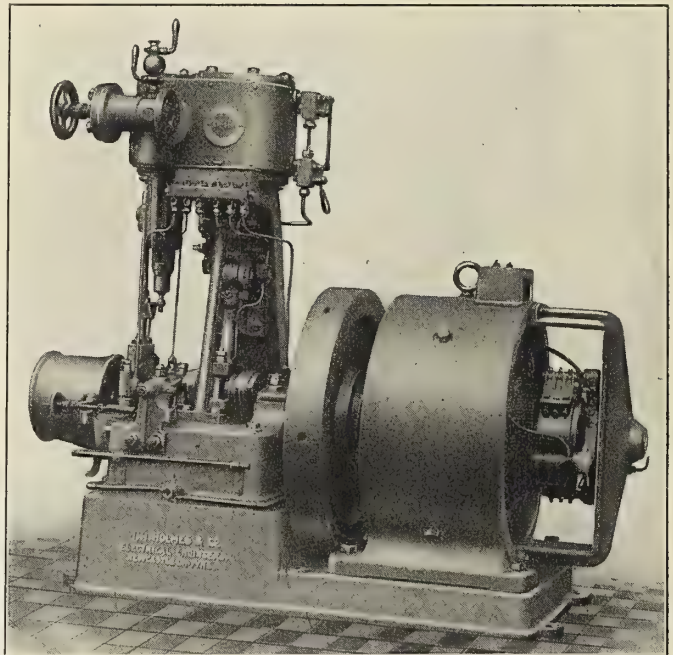
In drop forgings, vanadium steel flows quite readily in the die, its heat treatment being very simple. No more trouble is experienced in machining it than is met with in the ordinary

steel with an equal proportion of carbon. The forgings take a high finish. Its peculiar quality to resist deterioration, arising from vibration, strain and fatigue, is exemplified in the service of a large number of vanadium steel springs now in use. A locomotive tender truck spring exhibited had been straightened 10,000 times under much greater force than would have been necessary to flatten any other type of spring, and this without any permanent set. Among the other exhibits was a wheel taken from under a locomotive tender weighing 140,000 pounds, this wheel having run 26,875 miles and showing a wear of but $1/16$ inch in diameter. The average life of the regular steel wheel is only a little more than 10,000 miles.

A six-throw crank shaft for an automobile was another item. Still another exhibit was part of a torpedo, recently tested with gratifying results. Where great strength, toughness and power of resisting shock are desired, this material seems to have a large future before it.

A Multipolar Generator Set.

The Castle type dynamo set is made by J. H. Holmes & Company, Newcastle-on-Tyne, the dynamo being coupled direct to a vertical engine, both on the one bedplate, and especially designed for use on shipboard. The engine is of the slow-speed type, all wearing parts having large surfaces and being supplied with efficient lubrication, so that continuous running involves no risk of heated bearings. The moving parts are so balanced as to run without vibration or excessive wear.



Steam pressures up to 170 pounds are used. In many cases a simple engine is fitted, but in some cases compound engines are substituted, running at the same speed, but necessarily more expensive per installation.

The type illustrated is built in a large variety of sizes, ranging from 1.1 to 18.5 kilowatts. The range in speed is from 450 to 300 revolutions per minute, all being designed for 100 volts. The smallest in the regular list has a cylinder 4 inches in diameter by 3 inches stroke, and measures complete 4 feet 2 inches in length, 18 inches in width and 3 feet 4 inches in height. The largest engine listed has a 9-inch cylinder with 7-inch stroke, and measures complete 6 feet 8 inches in length, 2 feet 10 inches in width and 5 feet 9 inches in height.

TECHNICAL PUBLICATIONS.

Lloyd's Register of American Yachts. Size, 9 by 7 inches. Pages, 454. Colored plates (flags), 45. New York, 1908: Lloyd's Register of Shipping. Price, \$7.50 (30/-).

The sixth annual volume shows a material increase in size over last year, and has been thoroughly revised in all particulars, especially in those relating to the engines of the rapidly-growing fleet of cruising launches. There are listed a total of 3,670 yachts, both sail and power, owned in the United States, Canada and the West Indies, with a total of some 3,500 yacht owners. The color plates give 2,013 private signals of American yachtsmen, and the burgees of 365 yacht clubs.

One of the most interesting features of the book, as showing the growth of American yachting, is the list of the yacht clubs. The first American Yacht List, published in 1874 by the late Neils Olsen, listed a total of thirty-two yacht clubs, and the greatest number listed prior to the establishment of Lloyd's Register of American Yachts was about 170. Lloyd's club list has grown steadily since 1903, until it has now reached a total of 386 clubs distributed in all parts of the United States and British North America. Not a few of these clubs have been established during the past winter.

This great increase is made up in three ways: first, of yacht clubs established in new localities; second, of new clubs established to meet the recent growth of the sport in localities where many clubs already exist. The third class of clubs, a large one and distributed in all parts of the country, is made up of the so-called "power boat," "motor boat" and "launch" clubs, organized by men who have no special interest in the older forms of yachting, but are enthusiasts in the cause of the modern power boat.

British Engineering Standards Coded Lists. Vol. V. Size, 8½ by 11 inches. Pages, 411 + xxxix. Structural Steel for Shipbuilding and Marine Boilers. Steel Castings and Forgings for Marine Purposes. Marine Code compiled by James Adamson. London, 1908: Robert Atkinson. Price, 25s. net.

The British Standard Marine Code is designed for the use of shipowners, ships' officers, shipbuilders, marine engineers and all those handling marine material, whether afloat or ashore. The latest details in all the various matters coming within the scope of modern marine engineering are carefully and systematically dealt with; and as the compiler has had a long experience of requirements for ships under various circumstances, he has incorporated the results in a most convenient form for ready reference by every one whose duty or interest lies in the direction of shipping and ships, sea or river navigation.

The code is divided into sections, and the index itself is a useful compilation for general service by all those whose business is connected with water-borne traffic. The ship is dealt with in a great many varied circumstances of location, of suitability for cargoes, of damage and repairs, under conditions derived from experience, including dry-docking and surveying necessities or requirements. The machinery in all details of latest equipment of turbine, watertube boilers, forced draft, hydraulic, electrical, refrigerating, steering gear and deck machinery are treated fully for repairs in all cases where found necessary.

In addition to the propelling machinery, all other machines, such as evaporators, feed-water filters and feed-water heaters, are also treated fully, with phrases for reporting condition of machinery, details, repairs, or renewals required; also ordering phrases for material connected with boilers and machinery, whether propelling or auxiliary. The propeller is dealt with under every circumstance; and with the assistance of a convenient sketch, a shaft can be ordered by telegram in full detail, or dimensions confirmed. The turbine has a sec-

tion to itself, and is given in minute particulars for all classes of repairs. The diagram for the location of turbine units, together with the phrases, will be found most useful.

Besides the marine code, this volume contains coded list of the British standard sections, and specifications for structural steel for shipbuilding, for marine boilers, ingot steel forging and steel castings for marine purposes. The great value of a good index in saving time and temper is well known; this has evidently been kept in view by the compiler of the code.

Autogenous Welding of Metals. By L. L. Bernier, M. E. Size, 4½ by 6½ inches. Pages, 45; illustrations, 32. *The Boiler Maker*, New York, 1908. Price, \$1.00 (4/-).

This work gives a detailed description of the various means employed for obtaining high temperatures for the welding of metals. The most common forms of apparatus are thoroughly described, and an account is given of how this process has been applied commercially. The book gives a comparison as to cost, suitability for various kinds of work and ease of handling of the various types of blow-pipe burners in use. The three important burners described are the oxyhydric burner, using oxygen and hydrogen; the oxyacetylene burner, using oxygen and acetylene, and the oxygas burner, using oxygen and illuminating gas. Comparison is made in the case of portable welding apparatus between blow-pipes using dissolved acetylene and oxygen and hydrogen, and, in the case of stationary welding apparatus, between oxyacetylene blow-pipes, using acetylene supplied by a generator, and the oxygas blow-pipe. In both cases the subject is thoroughly discussed from the point of view of the comparative cost prices of the two systems and from the point of view of easiness in handling and application. Charts are given on which curves have been plotted showing the total cost per unit distance to weld various thicknesses of metal by each of these systems. The curves show the total cost, including gas and workmanship, and also the cost in gas alone.

The Log of the Blue Dragon. By C. C. Lynam, M. A. Size, 6½ by 9¾ inches. Many illustrations, including maps. Oxford, 1908: Slatter & Rose. Price, 6/- net (\$1.50 net).

The marine engineer has to take into consideration a large number of conditions not immediately concerned with the question of machinery, and in consequence has a closer interest in the sailing ship than would appear at first sight. For this reason we make no apology for referring briefly to a new (second) edition of this work, which is a record of numerous cruises in a small boat the skipper of which is a well-known amateur sailor. His cruises range from Oxford on the River Thames, around the south coast of England and to the north of Scotland. Dangerous and difficult navigation is entailed in such a trip, and the logs are a record of great interest to anyone interested even remotely in the sea and in things marine. Those marine engineers, and they are many, who themselves regard sailing as a useful hobby, because it brings them into close touch with the conditions in which their inventions are tested, will find the book full of fascinating interest.

The Naval Pocketbook for 1908. Edited by G. S. L. Clowes. Size, 3½ by 5 inches. Pages, 991. Numerous illustrations. London, 1908: W. Thacker & Company, 2 Creed Lane, E. C. Price, 7/6 net (\$2.00).

This is the thirteenth year of a series of annual volumes dealing with the navies of the world, and giving much detailed information regarding the individual ships of which those navies are composed. These details include the usual dimensions, power, speed, battery and armor, and a considerable amount of miscellaneous items. There are also lists of the different types of guns used in various naval services, a complete table of the drydocks of the world and other subsidiary

tables. The last section includes outline sketches, showing battery and armor distribution of the principal ships of the various powers.

The work is decidedly conservative, in so far as information and illustrations are concerned, of the very latest ships laid down by Great Britain, Germany and Japan. It is true that extraordinary efforts are being made by these powers to keep the details from becoming public; at the same time much is known and much more is very closely surmised, so that sketches have been made representing probably the correct arrangement of the battery, and; to a certain extent, of the armor. This conservatism has led to the omission of sketches of such well-known ships as the British *Indomitable* class and the American *Delaware* class, in both of which the battery arrangements are thoroughly well known, and the armor also. In spite of these failings, however, the work is a very valuable compilation, and enjoys a deserved popularity. The number of ship illustrations is 127, representing probably upwards of 400 ships, because of the fact that very many illustrations cover several sister ships.

QUERIES AND ANSWERS.

Questions concerning marine engineering will be answered by the Editor in this column. Each communication must bear the name and address of the writer.

Q. 411.—Give a rule for figuring the pressure to be placed on a spring-loaded safety valve.

2.—What is the coal consumption of the *Lusitania* or *Mauretania* per voyage, per horsepower, etc.? J. S.

A.—The load to be placed on a spring-loaded safety valve would be equal in pounds to the area in square inches of the inside of the valve in communication with the steam inside the boiler, multiplied by the pressure in pounds per square inch at which it is desired to blow off. If the area of the valve is 6 square inches, and it is desired to blow off at 200 pounds per square inch, the spring should be so adjusted as to create a thrust against the valve seat of 1 200 pounds.

2. Regarding the coal consumption of the *Lusitania*, a paper read before the Institution of Naval Architects in April is reprinted on another page in this number, and gives the information desired.

Q. 412.—Regarding Lightship Number 88, at page 375, in your September, 1907, number, you give the beam as 29 feet; at page 197 in May, 1908, you give it as 27 feet. Which is correct? A. B. B.

A.—We are informed by the Lighthouse Board that the molded beam of this vessel is 29 feet.

Q. 413.—Please differentiate between the Clyde boiler and the Scotch boiler. G. W. T.

A.—The Scotch boiler is one having one or more internal flues or furnaces, with return tubes around and above these flues for the passage of the products of combustion to the up-take and funnel. The front end of the furnace is located at the front of the boiler. The rear end of the furnace opens into a combustion chamber, which connects the furnace with the tubes. The tubes, furnace and combustion chamber are entirely surrounded by water (except, of course, in the front end, where the furnace is arranged for firing and the tubes for discharging the products of combustion up the stack), this water being contained within the cylindrical shell of the boiler. This boiler is also known as the cylindrical marine boiler, and sometimes as the return tubular boiler.

The Clyde boiler is of the same general type, except that the combustion chamber is not surrounded by water. For this reason it is often called a "dry-back Scotch boiler." It is not used to any extent in marine service, and has in general only one furnace, where the Scotch boiler frequently has four.

The Scotch boiler is often made double-ended, each end being a duplicate of the other, and having usually either three or four furnaces in each end. In such case the combustion chambers for the two ends are sometimes merged into one

common chamber, but more often two chambers are used, separated by a water space with a thickness of 4 or 5 inches.

OBITUARY.

John B. Roach, president of the Delaware River Iron Shipbuilding & Engine Works, Chester, Pa., died suddenly June 16, at the age of 68. This yard attained prominence 25 years ago by constructing the first four steel vessels (the A, B, C and D) of the "new" United States navy—the *Atlanta*, *Boston*, *Chicago* and *Dolphin*. Among the later products of the yard are the two turbine steamers *Yale* and *Harvard*. The yard, under the management of Mr. Roach, and, previously, of his father, John Roach, attained an enviable reputation for careful and thorough workmanship.

SELECTED MARINE PATENTS.

The publication in this column of a patent specification does not necessarily imply editorial commendation.

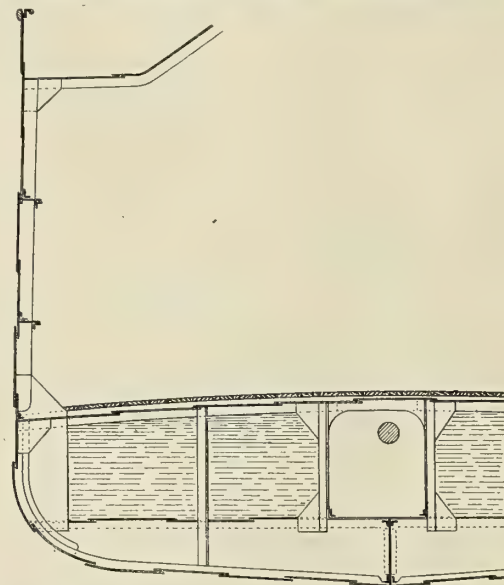
American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Loan & Trust Building, Washington, D. C.

884,474. RECIPROCATING PROPELLER. WILLIAM DAWSON, HAWARDEN, IOWA.

Claim 2.—A propeller comprising a plurality of tubular outer casings, having formed in their upper and lower sides longitudinally disposed guide channels, tubular inner casings adapted to fit and slide in said outer casings, guide studs arranged on the inner casings to engage the guide channels in said outer casings, whereby the inner casings are guided and steadied in their movement in said outer casings, means to reciprocate the inner casings, pivot shafts arranged therein, wings or blades pivotally mounted on said shafts and adapted to swing inwardly and outwardly to open and close the inner casings when same are reciprocated, and means to prevent said wings from folding entirely together when swung inwardly to open said casings. Three claims.

884,912. CONSTRUCTION OF SCREW-PROPELLED SHIPS. JOHN T. DUNCAN, CARDIFF.

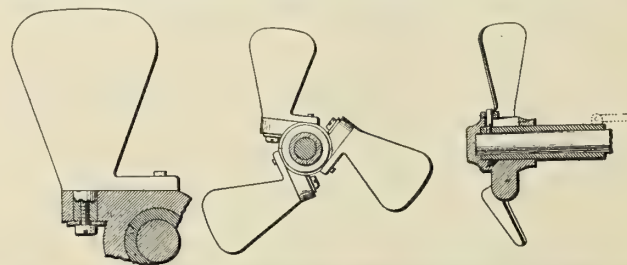
Claim 1.—In a screw-propelled ship, a shaft tunnel, a roof to said tunnel extending to the sides of the ship, and permanent water



ballast tanks on each side of said tunnel, located beneath the extended roof thereof. Five claims.

885,038. PROPELLER. ALBERT GNAEGY, CHESTER, ILL.

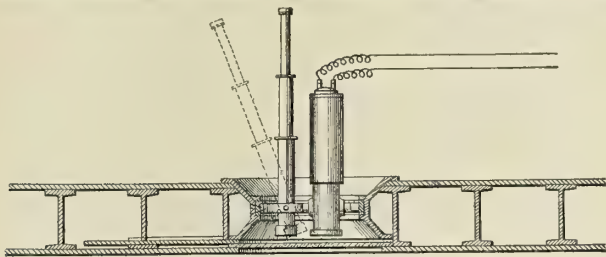
Abstract.—This invention relates particularly to that type of pro-



pellers having shiftable or reversible blades to facilitate the movement of a vessel in opposite directions. One claim.

885,087. SUBMARINE VIEWING APPARATUS. MANSOUR SAMAHA, WASHINGTON, D. C.

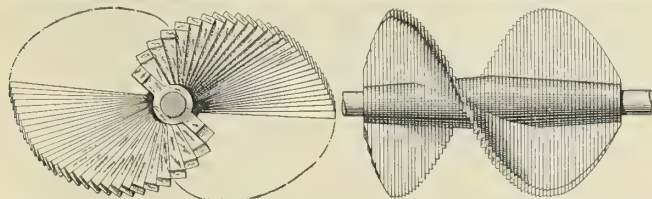
Claim 1.—The combination with a hull, of a submarine viewing apparatus comprising a single search light, and a plurality of fixed



optical tubes arranged at different distances from the axis of the search light, each having its axis arranged to intercept the axis of the search light at a predetermined distance from the hull. Fourteen claims.

885,109. SCREW PROPELLER. THOMAS F. J. TRUSS, LONDON.

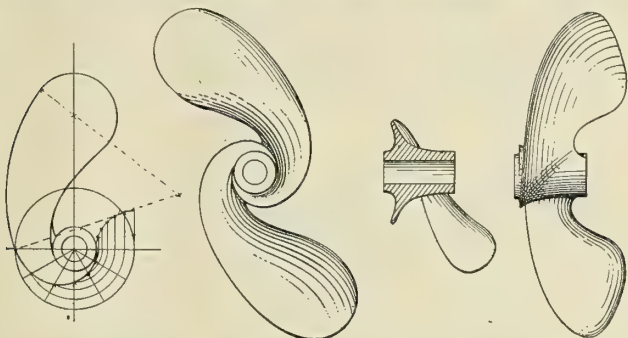
Claim 1.—In a screw propeller, the combination of an elongated boss tapered at the ends, and screw blades thereon of such a length



that there is no complete interval between the turns in end view, such boss and blades comprising a series of laminae each having arms, and central shaft-receiving boss portions secured together in such relation to one another as to form helicoidal blades, each lamina being tapered from the tip toward the boss. Six claims.

885,174. PROPELLER WHEEL. HARRY J. PERKINS, GRAND RAPIDS, MICH. ASSIGNOR OF ONE-THIRD TO FREDERICK L. PERKINS, AND ONE-THIRD TO CHARLES E. PERKINS, GRAND RAPIDS.

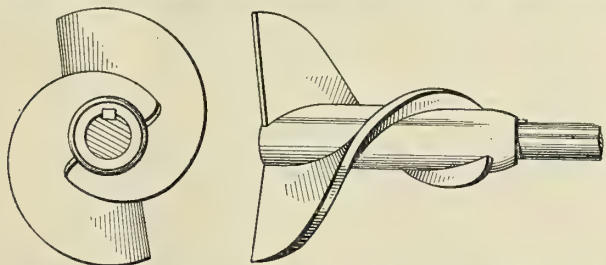
Claim 2.—A propeller wheel comprising a hub and a blade whose forward edge, when projected on a plane perpendicular to the axis,



commences tangential to the hub, thence extends spirally around the hub in a harmonic curve for 180 degrees, and thence is prolonged in a curve having a radius equal to the radius of the entire wheel, and with its center on a line struck from the outer end of said curve, and perpendicular to a line tangential to said curve at its outer end, said edge being curved rearward in substantially a cycloidal curve when projected on a plane parallel to the axis. Four claims.

885,250. PROPELLER. GEORGE W. HOPKINS, BROOKLYN, NEW YORK.

Claim.—A propeller comprising a central hub which is cylindrical in form in cross section, and provided with two spiral blades which



do not extend entirely around the hub, and the width of which increases rapidly from the front end to the rear end thereof, and the radial width of which is greatest at the rear end, where they terminate in straight lines which are oblique to the axis of the hub, and approximately tangential to the periphery thereof. One claim.

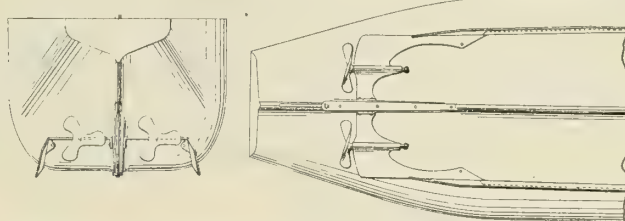
885,312. STEERING GEAR FOR BOATS. EARL C. AKERS, PORT HURON, MICH.

Claim 1.—A steering gear for boats comprising, in combination with the rudder and a rudder-actuating mechanism located abaft amidships and provided with a controlling device, a steering wheel located at the forward part of the boat, and a rigid inextensible connection

between said wheel and the controlling device of the rudder-actuating mechanism, arranged to operate the latter in both directions of movement of said connection, said connection serving to promptly transmit to the rudder the steering movements of the wheel, and operating to maintain the relation of the steering wheel to the rudder. Twenty-two claims.

885,370. HULL OF VESSELS. ISAAC E. PALMER, MIDDLETOWN, CONN.

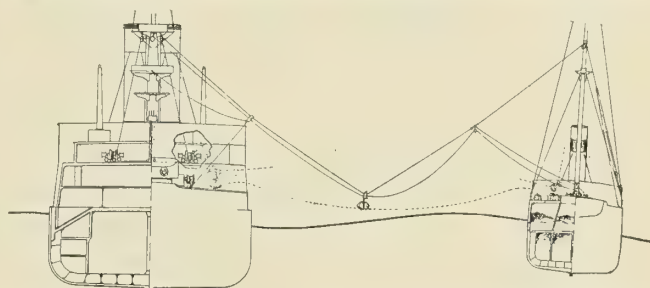
Claim 3.—The combination with the hull of a vessel provided with



a shaft bearing and bilge keels, of fins projecting in opposite directions from the shaft bearing, and forming an extension of the bilge keels. Three claims.

885,774. CONVEYING APPARATUS. THOMAS S. MILLER, SOUTH ORANGE, N. J.

Claim 13.—In combination, two boats advancing on courses in parallel, a cableway connecting the same containing a supporting rope and



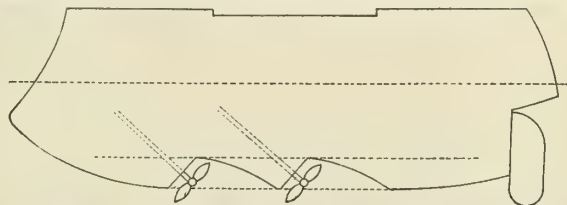
a load carriage traction rope, a take-up and pay-out mechanism on one of said boats exerting tension on said supporting rope, and a take-up and pay-out mechanism on the other of said boats exerting tension on said traction rope. Twenty-seven claims.

886,621. TRANSPORT VESSEL. OTTO MEHRTENS, KIEL, GERMANY. ASSIGNOR TO FRIED. KRUPP AKTIENGESELLSCHAFT GERMANIAWERFT, KIEL-GAARDEN.

Claim 1.—A transport vessel having a double hull extending throughout the bottom and up to the harbor deck, and a sea-proof superstructure surmounting the harbor deck, and having side walls extending directly from the inner wall of the hull and inclined to the horizontal at an angle approximately corresponding to the rake or slope of the cargo. Two claims.

886,850. PROPULSION OF VESSELS. GIOVANNI PERTOT, MILAN, ITALY.

Claim 2.—A navigable vessel having recesses in the bottom thereof on opposite sides of the keel at a distance from the bow equal to about one-



third the vessel's length, and propellers mounted on the rear ends of driving shafts, which are inclined to the horizontal in vertical planes parallel with the longitudinal axis of the vessel, the propellers turning in part within said recesses, but acting mainly on the outside fluid. Two claims.

British patents compiled by Edwards & Co., chartered patent agents and engineers, Chancery Lane Station Chambers, London, W. C.

25,896. BOLLARDS; CABLE-STOPPERS. F. S. PETT, DOVER.

Fairlead rollers are mounted on pins which pass through the bed-

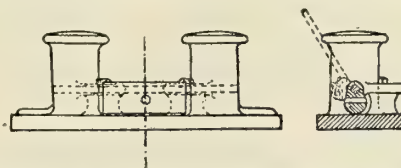


plate and a flange cast in one with the bollard. The rollers may be in any position adjacent to the bollard, and a cable stopper may be fitted between the bollards, for use in conjunction with the fairlead rollers.

26,132.—SEA-SOUNDING APPARATUS. W. THOMSON, KELVIN, BARON, F. W. CLARK, AND KELVIN & JAMES WHITE, GLASGOW.

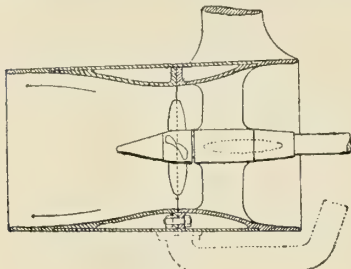
The machine is modified by mounting a motor below the reel to drive it through bevel-gearing for winding-in. A smooth-surfaced wheel facilitates handling when the wire is being rapidly wound in.

25,628. SALVAGING SHIPS. A. BECCHI, AND G. B. TARRANTINI, GENOA, ITALY.

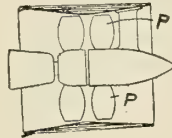
A self-propelled apparatus for burrowing underneath a sunken ship, carrying a rope with it, consists of a casing provided externally with two propellers or screw augers on shafts which are set at an angle to one another. The propellers are driven by a motor, which also actuates a pump used to empty the ballast tank. When the apparatus has reached the bottom of the ship, the intake valve is closed, and the ballast tank begins to empty.

27,133. SCREW PROPELLERS. Y. WADAGAKI, NODEO, KINOSAKI, JAPAN.

A fixed or hinged tube is situated around the screw propeller and is flared at its forward and after ends, the cross-sectional area decreasing from each end towards the mid-length or a point situated a little forward of the mid-length of the tube. The screw propeller is situated at the most contracted portion of the tube, so that it may operate in water having a relative velocity much higher than the forward speed of the vessel. The outlet area is slightly less than the inlet area. The tube may be in two parts, covered with plating, fitting flush with the outer surface, and it may be pivoted so that it may be turned to steer the vessel.



27,133.



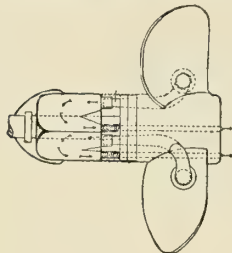
27,181.

27,181. SCREW PROPELLERS. A. RIGG, LONDON.

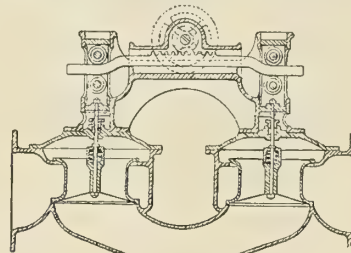
A screw propeller is mounted within a casing or tunnel which is fitted astern of the propeller, with fixed or adjustable guide blades, P. The casing is constructed with a bell-mouth forward, or forward and aft, the narrowest portion being situated between the propeller and the guide blades. The exterior of the casing is provided with a suitable filling, in order to present a smooth outer surface.

27,235. SCREW PROPELLERS; PROPELLING BY WATER-JETS. W. LOVIS, AND A. RANKIN, LONDON, E.

The blades of screw propellers are provided with perforations to which are attached tubes leading to the boss. Connected with the tubes are tubes telescoping in the casing, which is slidable on the shaft. Rotation of the propeller forces water through the tubes into the casing, from which it discharges, through pipes, to the after end of the propeller boss, to assist in propulsion.



27,235.



27,940.

27,940. ELASTIC FLUID TURBINES. E. BROWN, OF BROWN, BOVERI & COMPANY, BADEN, SWITZERLAND.

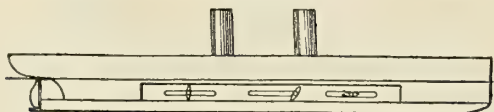
Relates to controlling valves for marine turbines which control the engines from a single point, and comprises means whereby some valves are positively held in the closed position while others are opened. The valves are actuated by a rack-driven cam-bar, which bears against rollers. In another form, the valves are operated by a worm-driven cam, which acts on pivoted levers. In both forms, a spring connection is provided between the valve and the spindle, to take up any slack.

27,409. ELASTIC FLUID TURBINES. R. PAWLKOWSKI, GORLITZ, GERMANY.

The loss of steam through the clearance spaces of a steam turbine is prevented by causing the steam to impinge upon guide and moving vanes which are not parallel, so that the steam passes along cylindrical surfaces concentric with the drum. In one form, this is accomplished by arranging straight vanes obliquely to the radii; in another form, the vanes are bent longitudinally so as to form involutes to the drum. A further method, applied to involute vanes having a portion bent backward, consists in twisting the vanes so that their axes lie obliquely to the generatrix of the cylinder. The invention may be applied to either the guide or moving vanes, or to both.

27,565. SHIPS. PROPELLING BY RECIPROCATING PLATES. J. CRAVEN, THORNTON, YORKSHIRE.

A number of plates are housed in recesses in the side of a ship,



and are connected to shafts, linked by arms to rollers, the shafts and rollers working in two guides. The first guide is a straight slot, and the shaft which slides in it is connected to the piston of the engine. The roller moves along the curved guide in such a manner that the

plate is vertical during one stroke, and horizontal during the return stroke. A modification is described in which the roller is replaced by a toothed wheel engaging a rack on the guide.

27,917. SHIPS' STEERING GEAR. VICKERS, SONS & MAXIM, BARROW, AND A. T. DAWSON, WESTMINSTER, LONDON.

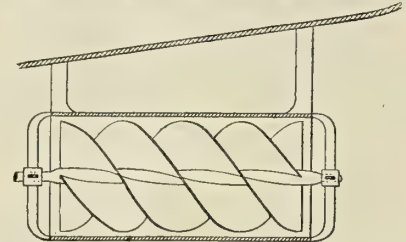
Arrangements are made to bring electric or other motive power automatically into action for augmenting or replacing the manual operation of the apparatus. The steering wheel is mounted on a shaft, which passes loosely through a sleeve having upon it a worm in gear with a pinion, loosely mounted on the vertical shaft. The pinion drives the bevel gears and rotates the vertical shaft, connected by bevel wheels to the horizontal shaft, by which the steering gear is operated.

28,367. SHIPS' CABINS. N. CHAMBERLAIN, J. STOKES AND HOSKINS & SON, NEPTUNE WORKS, BIRMINGHAM.

Portable rooms or cabins are constructed of sectional bulkheads or partitions, retained in position by cross-rails or their equivalent, detachably secured to stanchions fitted between the decks. The side of the cabin in which the door is situated is constructed of three sections. The edges of the central section, containing the door, are convex in form and fit into the concave edges of the other two sections. The remainder of the sections which form the cabin are similar to the latter two sections, so that they may be interchangeable. The pieces of each section are stiffened by bars of trough or channel section, and by the socket bars which carry the berths.

28,437. SCREW PROPELLERS. J. R. PORTER, MIDDLESEX.

One or more cylinders, which are wholly immersed, are fixed to the vessel by struts, and carry within them shafts upon which are



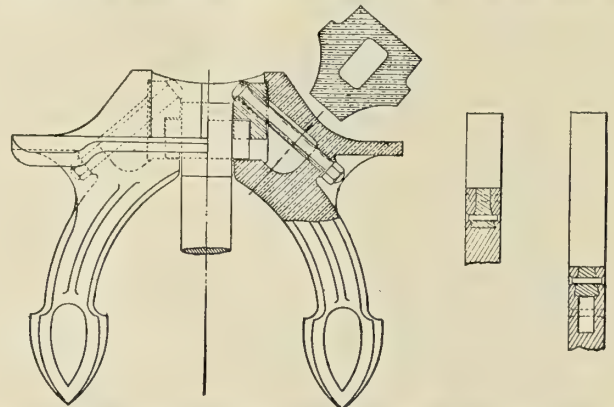
helical blades. The length of one convolution of the blades is the same as the length of the inclosing cylinder. The blades may be bent over so that their outer edges are tangential to the cylinder, and the extremities of the blades may be connected at intervals by hoops.

28,447. SHIPS' TANKS. I. P. B. KNUDSEN, COPENHAGEN, DENMARK.

In ships provided with top water-ballast tanks arranged along the sides, warm water from the condenser is pumped through these tanks, instead of steam, to prevent the water in them from freezing.

29,063. ANCHORS. G. HEPBURN, LIVERPOOL, AND J. E. FLETCHER, DUDLEY, WORCESTERSHIRE.

In stockless anchors of the type in which the shank is provided with trunnions, which fit in sockets in a cavity in the head, and



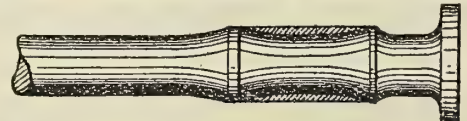
are held in place by blocks, the bolts securing the blocks in position are set obliquely, one end being in the crown of the anchor and the other between the trippers and the flukes. To insure good and sound castings, a cavity is cored out, and the metal of the crown is carried to the trippers in a plain flush surface. Two webs are provided when the flukes meet the head of the anchor, forming cavities which contain the nuts of the bolts.

29,407. ELASTIC FLUID TURBINES. MASCHINENFABRIK OERLIKON, ZURICH, SWITZERLAND.

Turbine blades are axially riveted in undercut grooves formed on the periphery of the rotor. The bottom of the undercut grooves may be provided with a second annular groove, into which the roots of the vanes or distance pieces may or may not extend.

29,442. PROPELLER SHAFTS. R. THOMPSON, WEST HARTLEPOOL.

The tail shaft is enlarged at the parts that adjoin the brass liners, and are most liable to corrosion, instead of being of uniform diameter



throughout. Collars are forged on the shaft, and the recessed surfaces between them accommodate brass liners, which are cast in place. The thinned-off edges near the collar insure a minimum of electric action, and consequently of corrosion.